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A Cable Forwarding System Proof of

Concept Facility Design

USDA-MSU Cooperative Research Project
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PROOF OF CONCEPT FACILITY DESIGN

USDA-MSU COOPERATIVE RESEARCH PROJECT 1984-1985
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USFS-MSU Cooperative Research Project

A cable Forwarding System - Proof of Concept Facility Design

Section 0. Introduction

Objective: To develop a test facility that can simulate field conditions for a cable forwarding system. The system must be able to duplicate expected loading conditions and evaluate system operating parameters. The design will consist of the following components.

Section 1. Cable segments, cable connectors, design speeds, loads and tension limits. Operating limits and operating equations.

Section 2. Sheave selection and sheave guides to pass cable connectors and chokers.

Section 3. Hydraulic motors, gear reduction and sheave bearing design.

Section 4. Sheave support frame and mounting method.

Section 5. Sheave support tension sensor. Try a hydraulic cylinder with pressure sensor and a strain gage sensor on the cylinder rod so the two methods can be compared.

Section 6. Select the angle sensors for the angle of the support frame to Ts (β) and the angle of the outgoing line (θ).

Section 7. Select the operating parameters for the cable speed sensor. This is the primary control variable and the system compares the desired speed with the actual speed for control change.

Section 8. Develop a plan for controlling hydraulic motor speed. Compare a needle valve with a spool type throttling valve for linearity and range of response.

Section 9. Develop a hydraulic motor displacement control. This uses a variable volume motor with an external lever control.

Section 10. Determine the location and connections for the necessary pressure relief valves.

Section 11. Determine the charge pump capacity needed for anti-cavitation control and circulation to the heat exchanger.

Section 12. Determine the filter and reservoir requirements.

Section 13. Determine the heat exchanger requirements for the system.

Section 14. Determine the alternator, regulator, battery and alternator drive needs.

Section 15. Determine the pump and engine needs for hydraulic power supply for typical load units and the main drive unit. Use an alternative of gasoline, diesel and electric drive. Electric motors will be the choice for the test facility but the engine information will help for scale up.

Section 16. Develop a pump displacement control for loading devices (pressure control) and the drive unit (speed control).

Section 17. Develop the loading device and load simulation method for the devices.

Section 18. Develop the end sheave plan including mounting and cable take up.

Section 19. Develop an idler sheave plan including mounting.

Section 20. Develop the main drive unit including sheave or sheaves, motor, pump with speed and torque control.

Section 21. Develop the log loading unit including choker design, load support platform, cable positioning technique, load positioning technique and choker attachment device.

Section 22. Develop the load unloading device including the choker disconnect technique and log handling method.

Section 23. Develop a cable tension control device that will compensate for changes in cable length during loading.

Section 24. Develop a device and operating method for adding or removing cable segments.

Section 25. Develop a cable support frame with positions for attaching the sheave support frames.

Section 26. Develop a tentative test procedure for evaluating system performance.

Section 27. Develop a data collection system for measuring system performance during the test sequences.

Section 28. Develop a data analysis technique for visually displaying performance parameters and comparing them with the design parameters.

Section 29. Determine a construction budget for the test facility.

Section 30. Determine a construction schedule for the test facility.

Section 31. Propose a test schedule for the forwarding system.

Attachment 1. Grant Application to the USDA Competitive Research Grants Program for Forest and Rangeland Renewable Resources in Fiscal Year 1985.

Attachment 2. Brochure, "Cable Conveyor Concept - A Research and Development Prospectus" by W. E. Larsen, M. J. Gonsior and W. R. Taylor, 1985.

Attachment 3. Flow Chart and Computer Program for Analysis of Multi-Span, Multi-Load Cable Systems Between Two Control Sheaves, M.J. Gonsior, USDA Forestry Sciences Laboratory, Bozeman, MT, 1986.

Attachment 4. Improved Theory for Static Analysis of Cable Systems, Draft Manuscript, M.J. Gonsior, USDA Forestry Sciences Laboratory, Bozeman, MT, 1986.

Section 1. Cable segments, cable connectors, design speeds, loads and tension limits. Operating limits and operating equations.

The cable segments will utilize 3/8" wire rope for 1/2 size scale and 1/4 strength scale factors. Each cable segment of 30 to 50 feet will be fitted with braided loop ends and heavy duty thimble inserts. The cable should operate at a design maximum tension of 5000 pounds or less. The design loads will be up to 500 pounds. The design cable speed will be the same as the design field speed and completely controlled from zero to 500 feet per minute. The grades simulated will go from zero to 60 per cent. The 60 per cent grade will require a braking or pulling force of 300 to 400 pounds for a fully suspended load. The pulling force could be nearer to 500 or 600 pounds, if the load is dragging with a very high coefficient of friction. It will be necessary to simulate a wide range of operating conditions to determine the ability of the system and its controls to respond to variable loads.

Two alternative connector links are proposed. The best connector will be the split cable connector shown in Figure 1.1. This cast link is one half the size proposed for the full scale system. If this cast link cannot be readily obtained, then a connecting link can be made from two C type missing link riveted chain connectors. The C link cable connector is shown in Figure 1.2. Insert the C links into a 2 1/2" diameter cylindrical section that will hold the sides of the C links so they will not spread apart. A solid spacer can be inserted between the C links to shorten the length of the connector. Two quarter-inch bolts through the cylindrical section and the C links will hold the components in place. The connector link will serve as a load attachment device as well as a cable connector to permit carrying the load without placing a concentrated load on the cable.

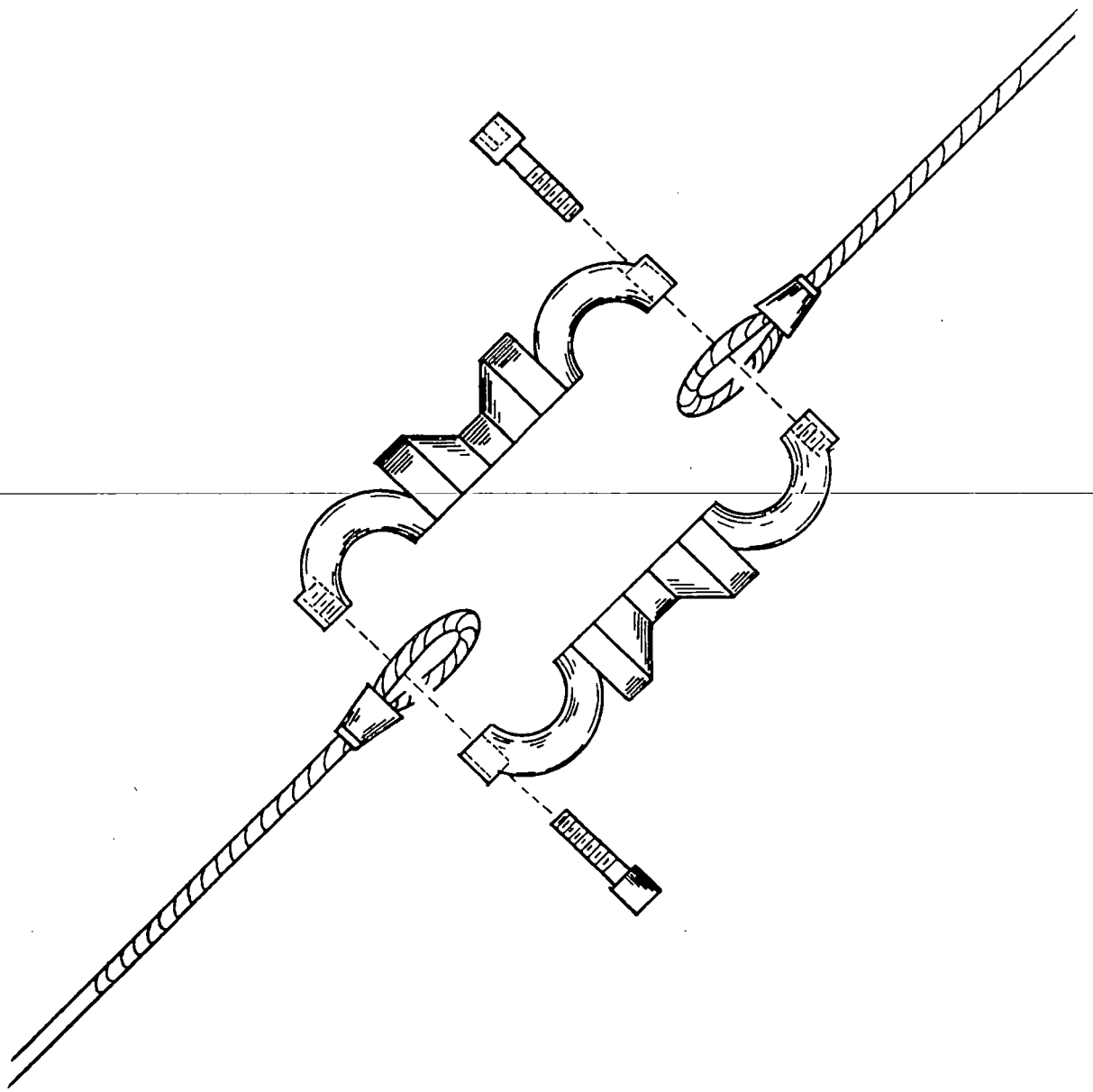


Figure 1.1 Split Cable Connector.

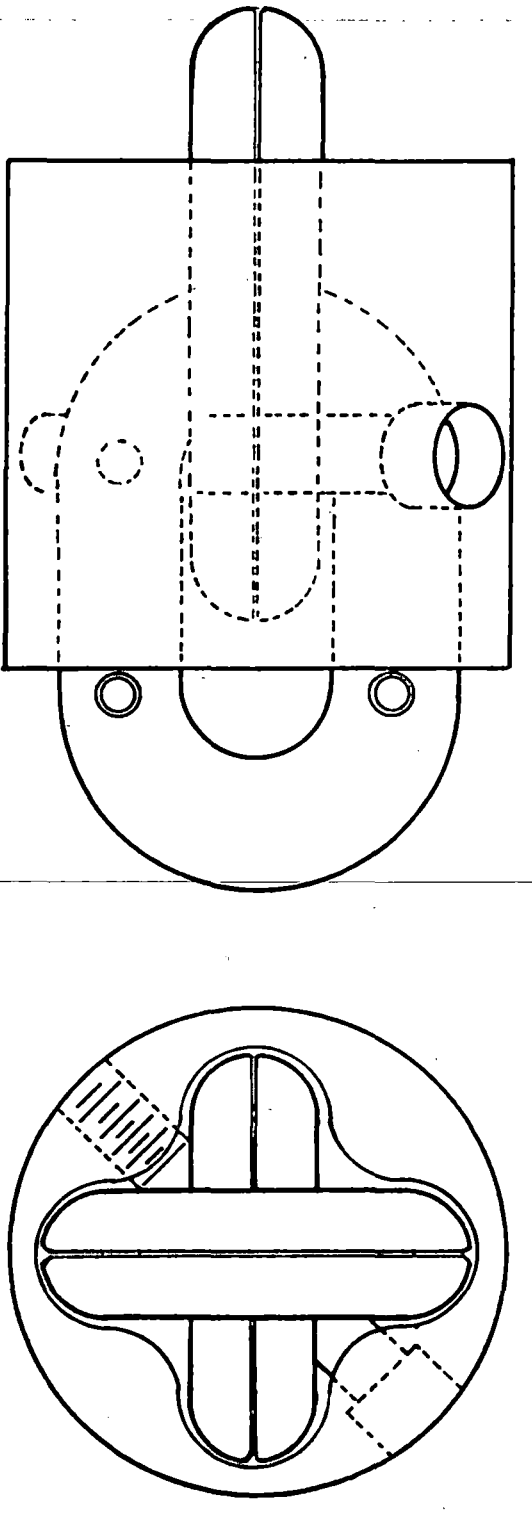


Figure 1.2 C Link Cable Connector

The cable sections will be connected by placing the 2 1/2 inch diameter retainer over one cable. The two cable ends are then connected by two missing link (C type) connectors. The length adjustment block is placed inside the two C links to shorten the connection by approximately 5/8 inch. The retainer is then slipped over the two missing links to hold them in place without riveting. Two 1/4 inch socket head bolts hold the retainer in place. A groove will be added around the center of the retainer to serve as an attachment point for the choker cable. This groove will not be as deep as the groove on the cast connectors and the cast connector will be shorter, therefore, probably better. The choker cable will be looped over the cable connector for carrying the load.

The operating equations for each control sheave are shown for equilibrium conditions in figure 1.3. These are based on the simplifying assumption that the sheave and its attachments are weightless. If β, $\theta 1$ and Ts are measured, the equations can be solved for ΔT, T1, T2, $\theta 2$, α and ΔT max as shown by the following equations.

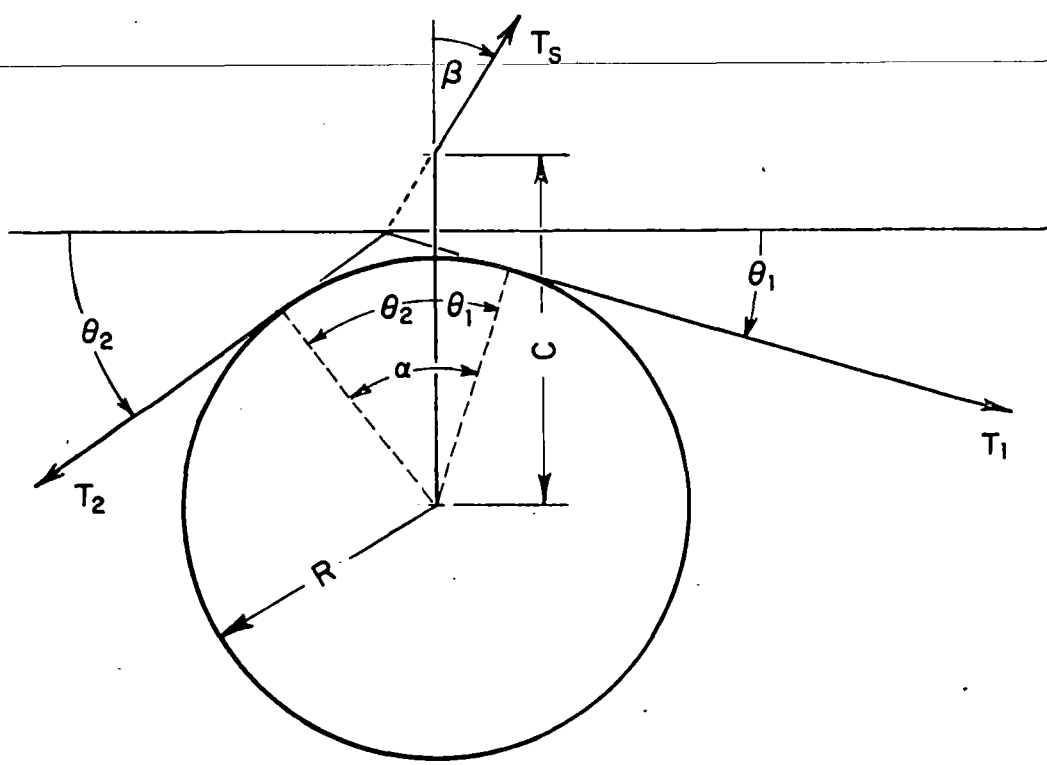


Figure 1.3 Equilibrium Conditions for the Control Sheave.

$\beta+$ in direction of cable travel

$\beta+$ = Pulling $\beta-$ = Braking

T1 in direction of cable travel - outbound from the sheave

$\theta 1$ measured on T1

T2 inbound to the sheave

$\theta 2$ calculated on T2

$\alpha = \theta 1 + \theta 2$ = Cable Wrap

1. $T2 \cos \theta 2 = T1 \cos \theta 1 + Ts \sin \beta$

2. $Ts \cos \beta = T1 \sin \theta 1 + T2 \sin \theta 2$

3. $\Delta T\ R = C\ Ts \sin \beta$

4. $\Delta T = T2 - T1$

Measure β, $\theta 1$ and Ts

Solve for T1, T2, $\theta 2$, α

1. $$\Delta T = \frac{C}{R} Ts \sin \beta$$

2. $$T1 = \frac{Ts^2 - \Delta T^2}{2\,(\Delta T + Ts(\sin \theta 1 \cos \beta - \cos \theta 1 \sin \beta))}$$

3. $T2 = T1 + \Delta T$

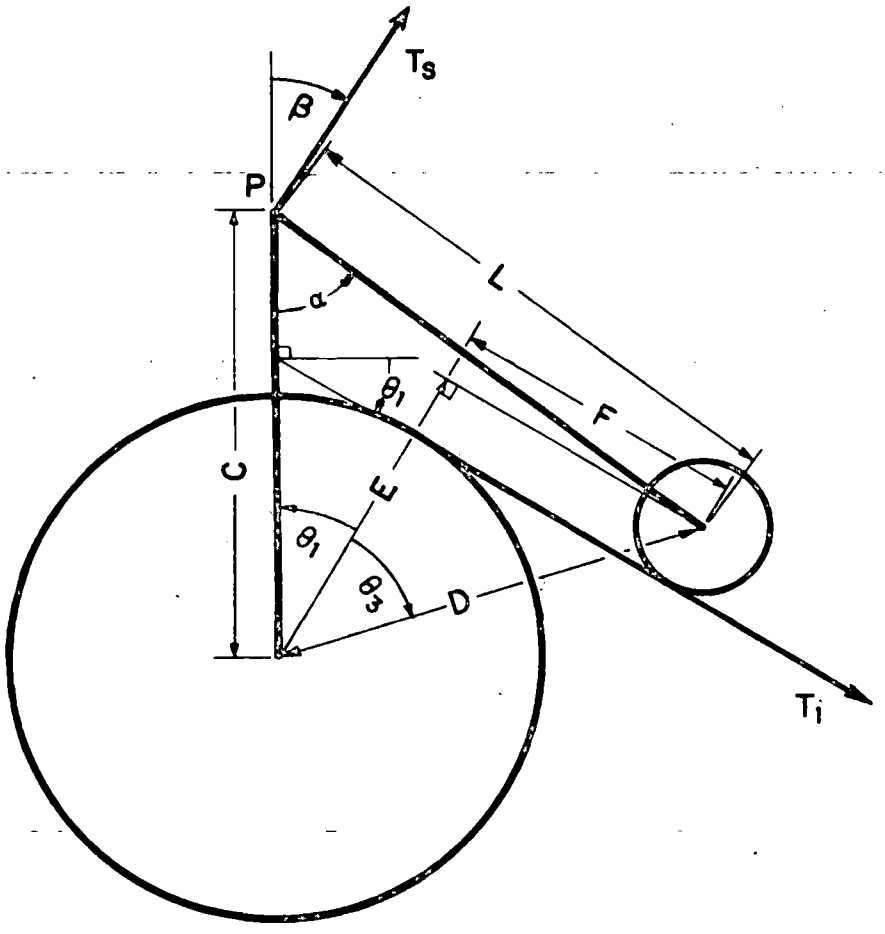
4. $$\theta 2 = \text{Arctan} \frac{Ts \cos \beta - T1 \sin \theta 1}{Ts \sin \beta + T1 \cos \theta 1}$$

5. $\alpha = \theta 1 + \theta 2$ = (wrap angle)

6. $\Delta T \max = T2\ (1 - e^{-c\alpha})$

c = coefficient of static friction between cable and sheave

The angle $\theta 1$, in Figure 1.3 could be measured from the hub of the sheave, however, an easier point will be from pivot point P shown in Figure 1.4.



Given: C, E, L

Measure: α

Calculate: $\theta 1$

Figure 1.4 Equilibrium Equations for Measuring the Cable Angle $\theta 1$.

1. $D^2 = C^2 + L^2 - 2\,C\,L\,\text{Cos}\,\alpha$

   $D = \sqrt{C^2 + L^2 - 2\,C\,L\,\text{Cos}\,\alpha}$

2. $F^2 = D^2 - E^2$

3. $L^2 = C^2 + D^2 - 2\,C\,D\,\text{Cos}\,(\theta 1 + \theta 3)$

   $$\text{Cos}\,(\theta 1 + \theta 3) = \frac{C^2 + D^2 - L^2}{2\,C\,D}$$

   $$\theta 1 + \theta 3 = \text{Arccos}\,\frac{C^2 + D^2 - L^2}{2\,C\,D}$$

4. $F^2 = E^2 + D^2 - 2\,E\,D\,\text{Cos}\,\theta 3$

   $\text{Cos}\,\theta 3 = (E^2 + D^2 - F^2)/(2\,E\,D)$

   $\theta 3 = \text{Arccos}\,((E^2 + D^2 - F^2)/(2\,E\,D))$

5. $\theta 1 = (\theta 1 + \theta 3) - \theta 3$

These equations can be simplified by eliminating D and F from the equation for $\theta 1$ as follows.

$$\theta 1 + \theta 3 = \text{Arccos}\ ((C^2 + D^2 - L^2)/(2\ C\ D))$$

$$D^2 = C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha$$

$$\theta 1 + \theta 3 = \frac{C^2 + C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha - L^2}{2\ C \sqrt{C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha}}$$

$$\theta 1 + \theta 3 = \text{Arccos}\ \frac{C - L\ \text{Cos}}{\sqrt{C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha}}$$

$$\theta 3 = \text{Arccos}\ \frac{E^2 + D^2 - F^2}{2\ E\ D}$$

$$D^2 = F^2 + E^2$$

$$\theta 3 = \text{Arccos}\ \frac{2\ E^2}{2\ E \sqrt{C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha}}$$

$$\theta 3 = \text{Arccos}\ (E / \sqrt{C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha})$$

Therefore,

$$\theta 1 = \text{Arccos}\ \frac{C - L\ \text{Cos}\ \alpha}{\sqrt{C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha}} - \text{Arccos}\ \frac{E}{\sqrt{C^2 + L^2 - 2\ C\ L\ \text{Cos}\ \alpha}}$$

Where:

C = distance from center of sheave to hinge point for measuring angle

L = Length of pivot arm to the center of the roller follower on the outbound cable

E = (Pitch radius of the sheave plus the pitch radius of the roller follower)

α = Angle between C and L (This is not the same as the wrap angle in Figure 1.3)

Monitor shaft speed - The control signal used for speed control is the difference between design speed and the actual speed. The master control determines the design speed for all units. The cable speed will vary from 0 to 500 ft/min. Shaft speed will probably be 0 to 150 RPM. The feedback signal for stability will be acceleration.

The four operating conditions are:

1. Hold speed
2. Accelerate
3. Decelerate
4. Hold stopped

The four limits on operation are:

1. T2 > Tmax, pulling too hard - decelerate
2. T1 > Tmax, braking too hard - accelerate
3. ΔT > Tmax pulling - decelerate, braking - accelerate
4. T1 and T2 > Tmax, help, upline - accelerate, downline - decelerate
5. Speed too great, help, upline, downline - decelerate

Each control system will contain a fixed displacement motor on the incoming line and a variable displacement motor on the outgoing line. Most of the control systems will be powered by the weight of the load on the cable. Some of the systems may also contain a variable displacement pump with an external power source. The fixed displacement motor will be mounted in series with the variable displacement motor and act as a pump for the variable displacement motor. A variable displacement flow control will be located after the variable displacement motor as a method for absorbing excess energy in a braking mode. The control functions for these units are as follows:

A. Fixed displacement motor - no external power

1. Hold speed - no change in the flow control
2. Accelerate - open the flow control
3. Decelerate - close the flow control
4. Hold stopped - closed flow control

B. Variable displacement motor - no external power

1. Hold speed - no change in motor displacement
2. Accelerate - decrease motor displacement
3. Decelerate - increase motor displacement

   If the input pressure approaches the charge pressure then close the flow control.

4. Hold stopped - closed flow control - displacement equal to the fixed displacement motor (A).

C. Fixed displacement motor - external power supply with variable displacement pump.

1. Hold speed - no change in flow control or pump displacement
2. Accelerate - open flow control then increase pump displacement. When the flow control is fully open or when the pump pressure approaches the charge pressure, then close the flow control.
3. Decelerate - Decrease pump displacement.

   If pump pressure approaches the charge pressure, then close the flow control.

4. Hold stopped - closed flow control and pump displacement moved to zero.

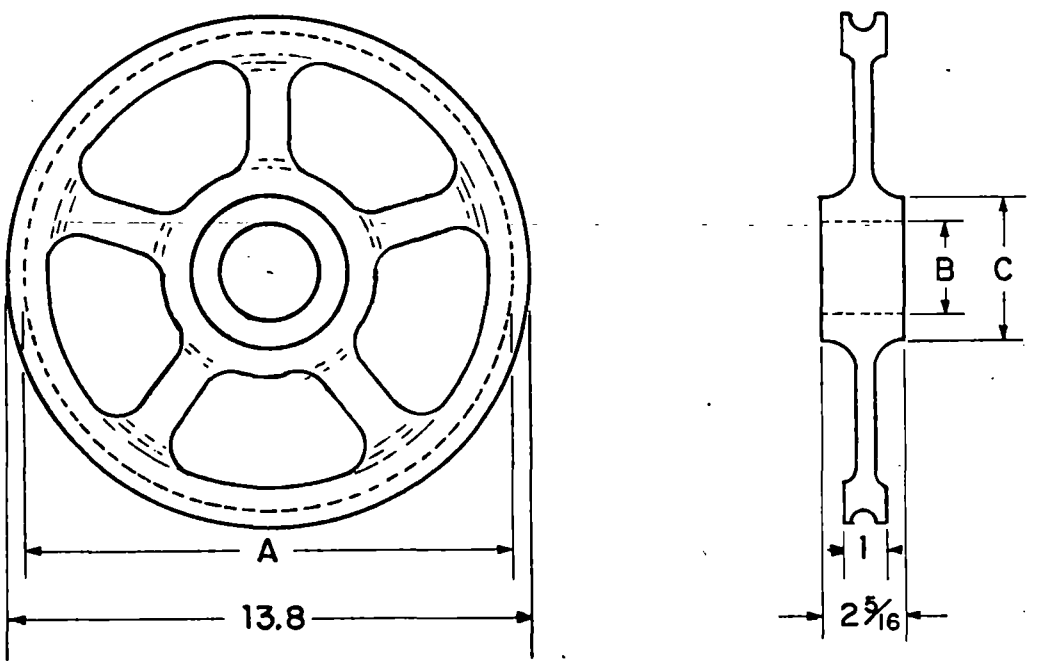
D. Variable displacement motor - external power supply with variable displacement pump.

1. Hold speed - no change in motor displacement or pump displacement.
2. Accelerate - decrease motor displacement.
3. Decelerate - Increase motor displacement

   If the input pressure to the motor approaches the charge pressure then close the flow control.
4. Hold stopped - closed flow control - motor displacement equal to the fixed motor displacement as shown in C.

Section 2. Sheave selection and sheave guides to pass the cable connectors and chokers.

The sheaves should have a minimum diameter of 30 times the cable diameter to avoid concentrated loads due to bending around the sheave. The sheave selected for the test facility is a McKissic 14-8 with the dimensions shown in Figure 2.1.



A. 12.72 to bottom of groove 13.1 pitch diameter with 3/8 cable

B. 3 inch finished bore or 2 1/2 inch bronze bushed bearing

C. Unfinished hub 3.83, could be turned to 3.75 inches.

Figure 2.1 Sheave Dimensions.

The bushed sheave can be used for non-powered applications such as idlers and the end wheel. The finished bore will be used with all powered applications. The 3/8 wall in the hub is too light for a 3/4 inch key which is standard for a 3 inch shaft. This particular application has high radial loads from the cable tension and very low torque loads due to the braking or pulling forces. A 1/4 inch key will be adequate for the torque loads.

The sheave guides will be assembled from 1/4 inch steel plate. Turn the center of a 13 7/8 to 14 inch diameter steel plate to 3.75 inches to fit the turned hub on the sheave. Form two steel hoops from 3/4 pipe or 1" diameter rod with a hoop outside diameter of 14.8 to 14.9 inches to serve as a smooth surface for carrying choker cables over the edge of the sheave. Weld the steel hoop to the circular plate as shown in Figure 2.2. Attach plates to both sides of the sheaves with 6 to 12 bolts 3/16 inches in diameter. Align bolts near the wheel spokes to prevent rotation of the guides on the sheaves.

A free wheeling star wheel retainer will be added to the outside of the sheave to hold the load on the sheave as the connecting link and the choker pass through. The detail for the star wheel is shown in Figure 19.2. The star wheel retainer will be mounted on the sheave hub but will be free to rotate relative to the sheave. Loads on the star wheel will be transmitted to the sheave through a self aligning bearing.

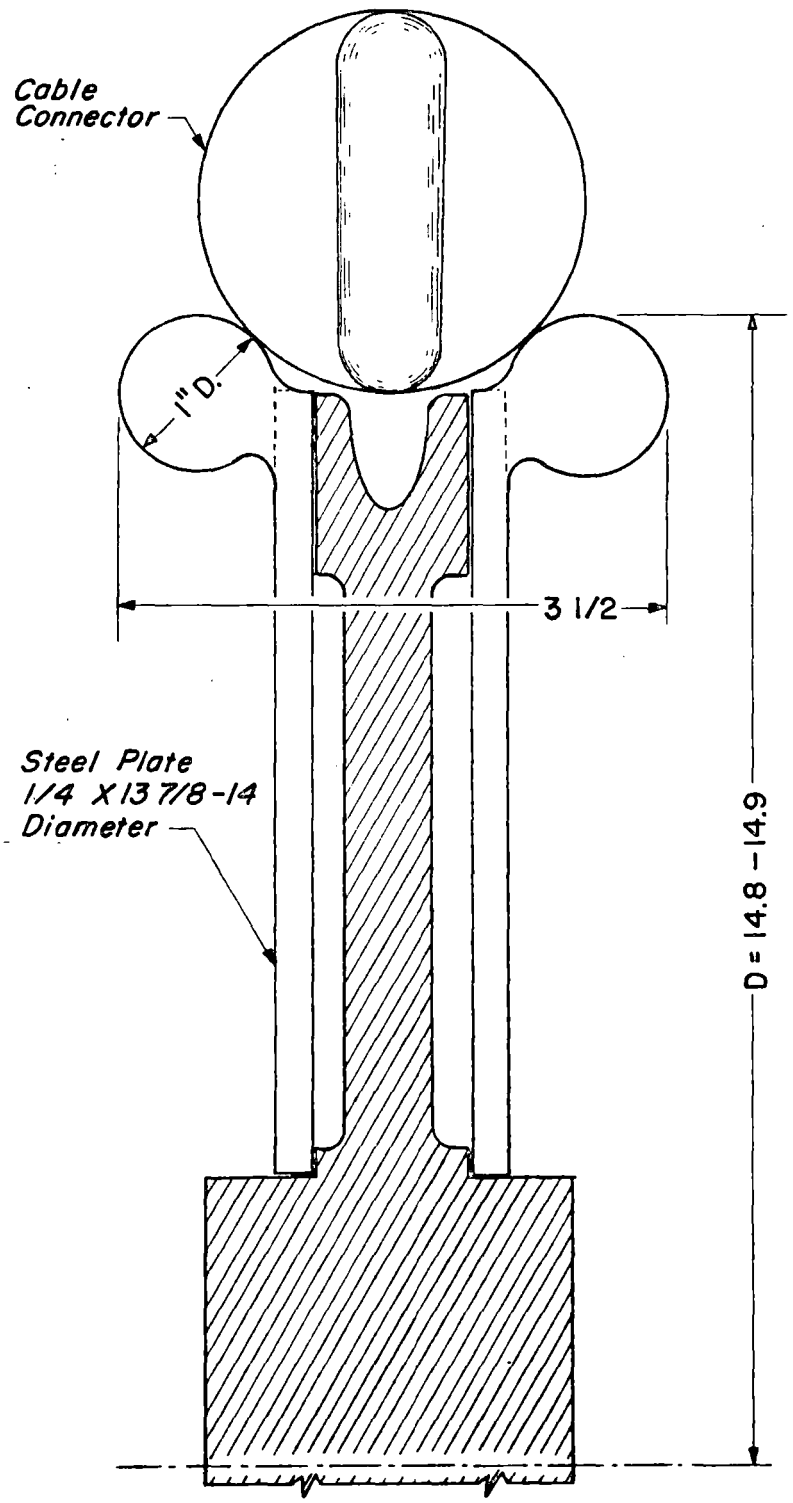


Figure 2.2 Guides for Cable Connectors.

Section 3. Hydraulic motors, gear reduction and sheave bearing design.

The cable will have a maximum speed of 500 feet per minute and the tension difference required to move a load from one tower to the next will vary with grade. At the steepest grade of 60% the tension difference with full suspension will be as much as 300 to 400 pounds for a 500 to 600 pound load. As the cable tensions are decreased the tension difference will go still higher, therefore, a design maximum tension difference of 500 pounds was assumed. The cable power at this tension difference is:

$$\text{Power} = \frac{500 \text{ ft/min} \times 500 \text{ pounds}}{33{,}000 \text{ ft pounds/min/HP}} = 7.6 \text{ HP}$$

The cable wrap required to develop this amount of tension difference is determined by the formula:

$$\Delta T \text{ max} = T2 \, (1 - e^{-c\alpha})$$

In this equation ΔT max is the maximum tension difference that can be carried before slippage occurs, T2 is the tight side tension on the cable and c is the coefficient of friction between the cable and the sheave. The angle α is the cable wrap around the sheave in radians. The equation can be rewritten as

$$\frac{\Delta T\text{max}}{T2} = (1 - e^{-c\alpha})$$

A graph of this equation is shown in Figure 3.1. At 20 degrees wrap angle and a coefficient of friction of 0.3 the value of ΔT max/T2 is approximately 0.1. If we reach ΔT max of 500 pounds then T2 must be equal to 5000 pounds. This is the design limit for cable tension; therefore, we will have to maintain the maximum cable tension to obtain this design maximum tension difference when the wrap angle is 20 degrees. With a cable tension of 4000 pounds then the ratio of ΔT max/T2 is 500/4000 or 0.125. This will require a wrap angle of 25 degrees or more if c = 0.3. This appears to be a practical minimum design wrap angle. Approximately 8 degrees of this wrap can be obtained by the weight of

the cable and the weight of the load. The remaining 17 or more degrees will need to be obtained by the zig-zag layout or elevation changes.

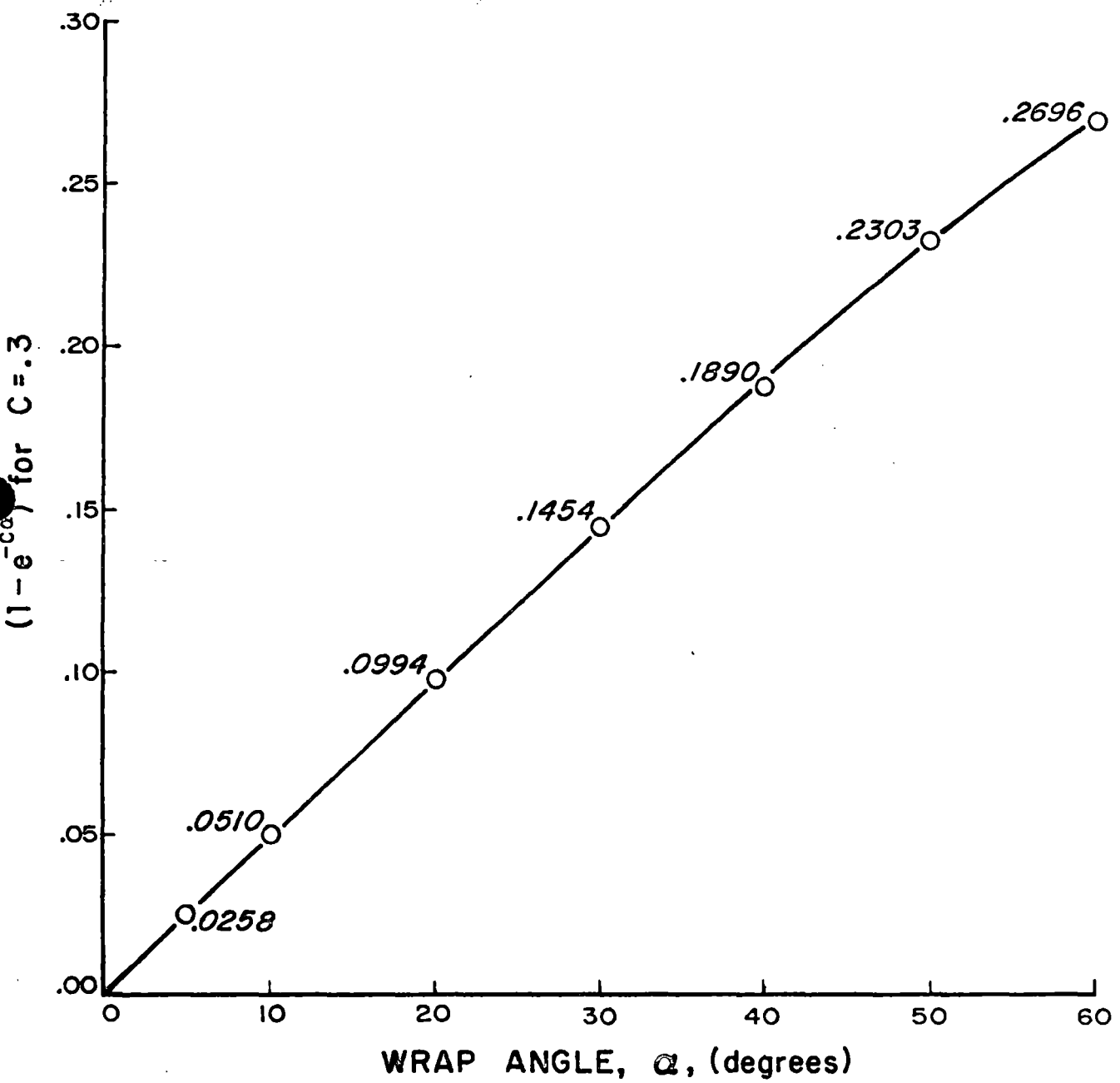


Figure 3.1 The effect of wrap angle on the ratio of max tension difference to the tight side tension.

The hydraulic motors used to power or absorb the load must be able to handle the power developed in the cable. The motor efficiency is the ratio of mechanical power to hydraulic power. This will be greater than one for braking and less than one when power is applied through the motor.

The cable speed will vary from zero to 500 ft/min. The sheave pitch diameter is 13.1 inches giving an effective circumference of 3.43 feet per revolution. The sheave speed will, therefore, vary from zero to 146 RPM. This is very slow for a hydraulic motor so a gear box with a 5 to 1 ratio was selected to give a motor speed of zero to 730 RPM. At 100% efficiency and a design hydraulic pressure of 2500 psi, this would require a motor displacement of:

$d\ in^3/rev \times 730\ rev/min \times 2500\ lb/in^2 = 500\ lb \times 500\ ft/min \times 12\ in/ft.$

$d = 1.64\ in^3/rev$

Hydraulic motors can be obtained that withstand intermittent pressures of 3000 psi or greater.

Therefore, a design of 2500 psi with pressure relief settings of 3000 psi will allow 20% overload to compensate for motor efficiency. In this case the motor will handle the design loads in a power mode and will handle an overload of 40% in braking mode.

The variable displacement motor should be slightly larger than the fixed displacement motor so that it can operate at speeds slower as well as faster than the fixed displacement motor. A displacement of approximately 1.8 $in^3/rev.$ will meet this need.

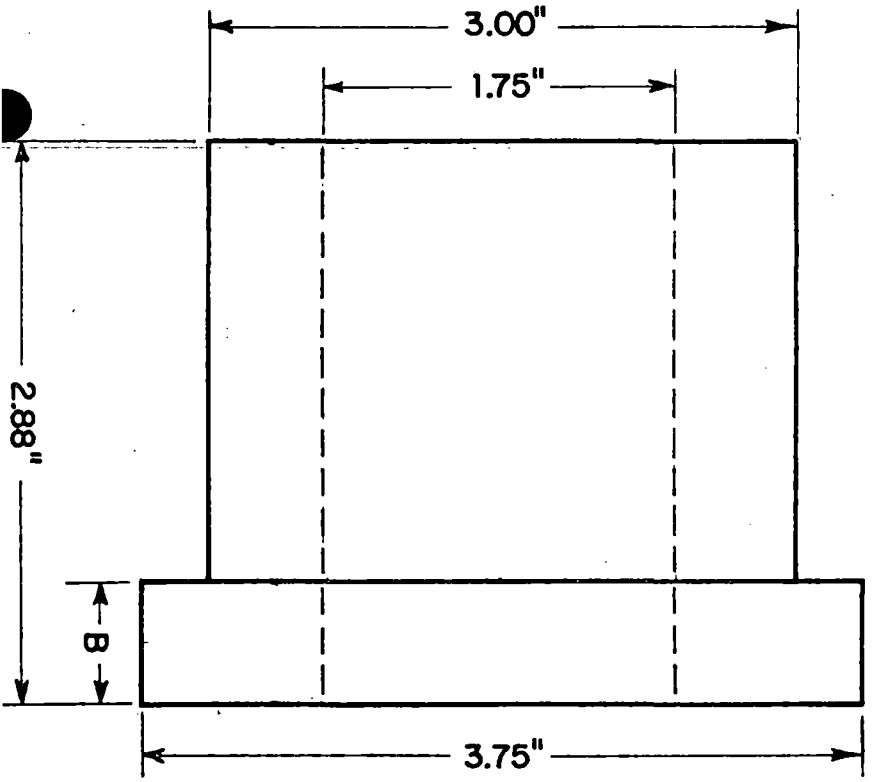
If the same gear ratio is used for the full scale system, then the sheave RPM will be reduced by 1/2 and the work increased by 4 times. This will then require motors with 4/.5 = 8 times greater displacement. An alternative would be to increase the gear ratio. A second stage speed increase with a 5 to 1 ratio will give an overall speed change of 25 to 1. This will require a motor

with 4/(.5 x 5) = 1.6 times the displacement. The resulting motor speed of 3650 RPM is faster than desirable for most motors. Therefore, a lower ratio and larger displacement will probably be desirable. The cost of large displacement motors will need to be compared with the cost of an additional gear reducer for final design.

A Terrell HD1-Style CB speed reducer was selected for this application. This gear box is designed for use with hydraulic motors and the motor will bolt directly to the gear box using standard SAE hydraulic motor and pump flanges. The gear box is relatively narrow (3.94 inches) and has an extended low speed shaft for mounting of the sheave. The gear box support includes four 1/2 inch bolts on 7.08 inch spacings. The power rating of this gear box is considerably greater than needed but the shaft size can be readily adapted to the sheave. It may be possible to utilize the same gear box in the large scale system, however, the output speed may be a little low. The same manufacturer makes a similar gear box with a heavier rating that would fall within the desired range. (Terrell HF1 - Speed Reducer)

The output shaft on the Terrell HD1 Style CB speed reducer is 1.75 inch diameter and 2.88 inches long. The sheave hub will need to be bushed from 1.75 to 3 inches. This bushing will have a 1/4 x 1/8 keyway in the outer surface for the sheave and a 3/8 x 3/16 inch keyway for the shaft.

Drill and tap the end of the shaft then use a 3.75 diameter by 3/8 thick washer to hold the sheave and hub on the shaft.



B. Collar 1/2 to 5/8 as needed to cause sheave guide to clear gear box (.57 best to fit shaft)

Figure 3.2 Sheave Bushing Dimensions.

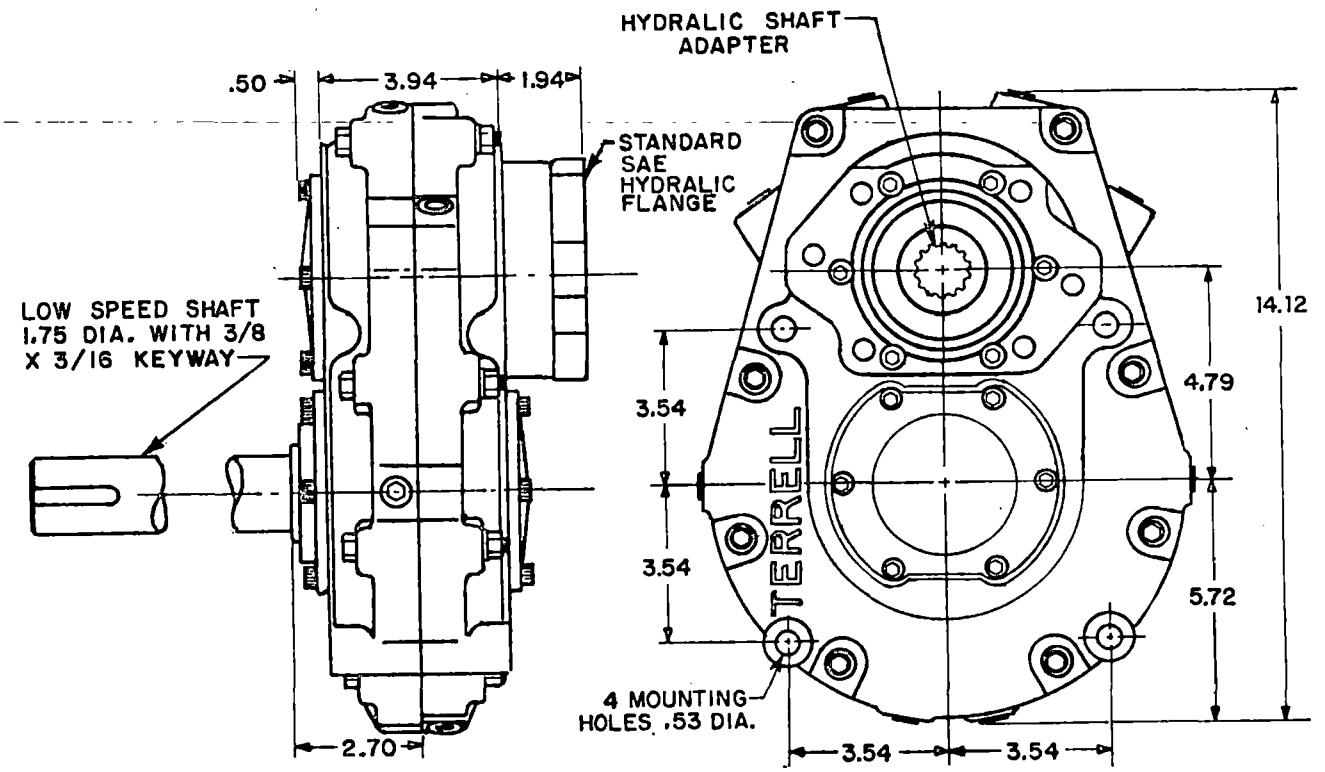
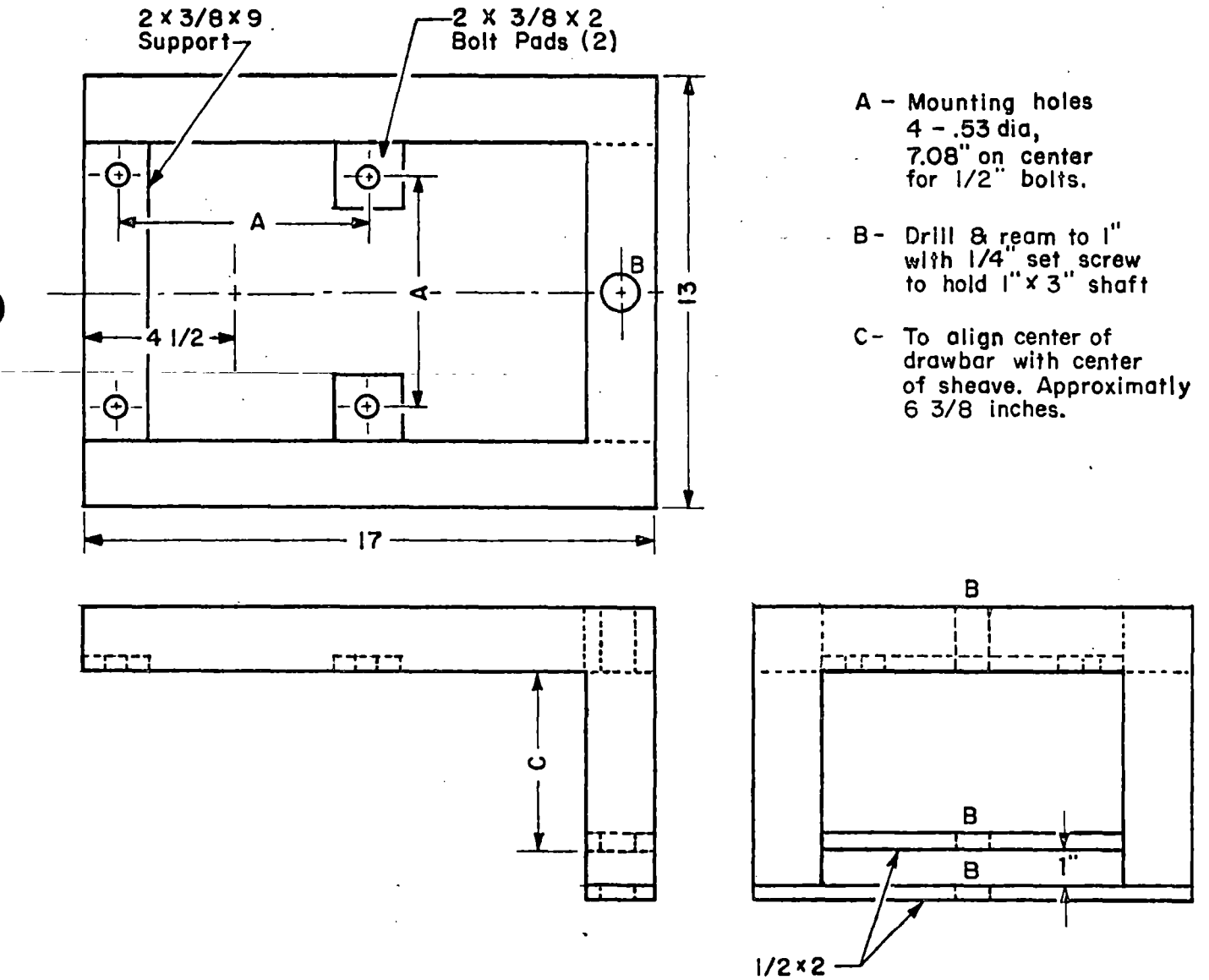


Figure 3.3 Terrell Model HD1 - Style CB Low Speed Shaft Extended.

Section 4. Sheave support frame and mounting method.

The sheave support frame is designed to attach the gear box to the load measuring transducer. The mounting holes on the gear case are in a square pattern 7.08 inches on each side with the output shaft in the center of the square. It is advantageous to keep the support frame fairly short to minimize the distance from the support structure to the cable. The connector to the support should be a clevis with a 1 inch pin. The pin should be locked with a set screw to permit its use as a reference for measuring angles. The support sensor and the cable angle sensor will fit over this pin in a clevis form.



A - Mounting holes 4 - .53 dia, 7.08" on center for 1/2" bolts.

B - Drill & ream to 1" with 1/4" set screw to hold 1" x 3" shaft

C - To align center of drawbar with center of sheave. Approximatly 6 3/8 inches.

Figure 4.1 Sheave Support Frame.

The main frame will be constructed from 2 x 2 x 3/16 tubing. The clevis portion will be made from two pieces of 1/2 X 2 bar.

The sheave support frame shown will allow approximately 5/8 inch clearance between the cable connector links and the end of a 1 x 4 drawbar. This will result in a distance of 11 1/2 inches between the center of the sheave and the center of the drawbar pin (called C in Figure 1.2).

Torque = $\Delta T\ R = C\ Ts\ \sin\beta$

ΔT has a maximum design value of 500 lbs.

C = 11 1/2 inches

R = 6.55 inches (the pitch radius of the sheave)

Ts must be approximated for a specific condition

$\sin\beta$ is the measured angle between the sheave support frame and the load tension sensor. This angle should vary from zero to approximately 30° to be in a nearly linear portion of the $\sin\beta$ curve.

Assuming a minimum wrap angle of 20° and a cable tension of 4000 to 5000 pounds. Then the support tension Ts can be approximated as:

$Ts = 2 \times (4000 \text{ to } 5000) \sin 10^\circ / \cos\beta$

$$\Delta T\ R = \frac{C \times 2\ (4000 \text{ to } 5000)\ \sin 10^\circ \times \sin\beta}{\cos\beta}$$

$$\tan\beta = \frac{\Delta T\ R}{C \times 2\ (4000 \text{ to } 5000)\ \sin 10^\circ} = (.173 \text{ to } .216)$$

$\beta = 9.8^\circ$ to 12.2°

As the angle of wrap increases the angle β will decrease. This will give a very small angle change in β. The change will be linear but will have to be measured accurately. The value of C used is this example is the shortest length for the frame that is practical. It provides clearance for the cable connector links and still applies a direct line of force from the support device through the plane of the cables.

Section 5. Sheave support sensor.

The sheave support sensor will be the connecting link between the load support frame and the support tower. This sensing unit needs to be accurately made and calibrated as the force measured by this sensor is used to determine cable tensions and torque. A number of different sensors are proposed and at least two should probably be tested. The loads on the sensor will depend on the cable tension and the amount of cable wrap around the sheave. With a cable tension of 5000 pounds and a wrap of 20° the load will be approximately 2000 pounds. With the same tension but a wrap of 180° (as would occur at the drive unit and elsewhere in the system) the support load would be 10,000 pounds. Therefore, the sensor needs to be designed for loads up to 10,000 pounds but with a sensitivity that can measure differences as small as 50 pounds. Strain gages and LVDT linear displacement devices are very sensitive. The strain gage units may be difficult to zero as it will be impractical to unload the transducer on a regular basis for zero calibration. The LVDT is extremely accurate and will have good zero stability but may not be very durable under difficult field conditions of vibration, dirt and moisture. In addition, it requires an AC power supply which would probably require a more complex instrumentation package. Linear potentiometers are simple and easy to zero as they are basically voltage dividers and it will be fairly easy to obtain a precision voltage source. A spring would be needed to carry the load and the pot would measure change of length in the spring. The pot uses a mechanical wiper which may wear with vibrations and may lose sensitivity with time.

The six load transducers under consideration are:

1. Hydraulic cylinder with pressure transducer.
2. Steel rod with strain gage transducer.
3. Aluminum rod with strain gage transducer.
4. Aluminum or steel drawbar with bored hole and strain gage transducer

5. Compression spring with linear transducer.

6. Load ring with linear transducer

1. A hydraulic cylinder with a pressure transducer.

This would be the cheapest transducer as the components are readily available as commercial units. A 2 1/2" diameter cylinder with a 1 inch rod will have an effective area of 4.12 square inches. With a 10,000 pound load this will give a pressure of 2415 psi. Cylinders and pressure transducers are readily available in this range. Most of the transducers are strain-gage type with a four arm bridge in the pressure transducer.

The disadvantage of a hydraulic cylinder are:

a. The length of the most common cylinder is 20 1/4 to 28 1/4 inches. (This is a standard 8 inch stroke cylinder with 20 1/4 inch retracted length). This could be reduced by 4 to 10 inches, but this would require considerable shop time.

b. There is some friction between the oil seals and the sides of the cylinder and rod. This is quite variable but could easily show as a hysteresis of $\pm$ 20 to 30 pounds.

c. There will be some oil leakage from the cylinder requiring a make up system to maintain the operating position of the piston. This would be no problem in the test facility as the cylinders could be pumped up and checked with a hand pump on a daily basis. An automatic system would be needed for field use.

d. It will be difficult but not impossible to unload the pressure transducer for determination of the zero balance position.

2. Steel rod with strain gage transducer.

This unit would be fairly easy to fabricate by utilizing cold rolled or ground steel shafting as the measuring unit. The ends could be machined to fit

standard hydraulic cylinder ends. Four strain gages would need to be cemented to the shaft, wired in the appropriate bridge form then weather proofed. The finished unit would have a center to center distance of 8 to 10 inches and would be very sensitive with no hysteresis.

The disadvantages of the steel rod transducer are:

a. It will be very difficult to unload the transducer for zero calibration.

b. The measuring sensor cannot be replaced under field conditions, however, the entire transducers with wiring could be easily replaced, if the load is supported.

c. Good weather proofing may be difficult to maintain under field conditions.

3. Aluminum rod with strain gage transducer.

This unit is almost identical to the steel rod, however, it will be more sensitive to changes. The advantages and disadvantages are approximately the same, however, it may be more expensive to fabricate.

4. Aluminum or steel draw bar with bored hole.

This unit will consist of a 1 x 3 inch aluminum or steel bar approximately 8 inches long. A one inch hole will be drilled in each end for mounting and a center hole approximately 2 to 2 1/4 inches in diameter will be bored in the middle of the drawbar. Strain gages are then cemented at the high stress and low stress points in the hole then wired into a bridge. The hole is then water proofed and sealed with silicone rubber. The unit will be very durable and weather proof.

The disadvantages of this unit are:

a. The limits with zero balance for strain gages will still apply.

5. Compression spring with a linear transducer.

A heavy compression or tension spring can be placed between the load support frame and the support tower. Changes in the length of the spring will be proportional to changes in load. The sensing unit could be either a linear potentiometer or an LVDT. The linear potentiometer would probably be the more practical sensing unit.

The disadvantages of this unit are:

a. Springs capable of handling 10,000 pounds will be extremely bulky and heavy.

b. Tension springs are probably not available in this size range with end mounting rings. The sensitivity at low loads would probably be poor due primarily to end effect on the springs.

c. Compression springs can probably be obtained but they will probably be designed for special application such as automotive and will probably be non-linear.

d. Load sensing with an LVDT could easily be affected by vibration.

e. Load sensing with a linear pot could be affected by pot wear.

6. Load ring with linear transducer.

A load ring is a very sensitive measuring device that can be used with a linear displacement transducer or with strain gages attached at the primary strain points. They can be very accurately made and will give precise results.

Disadvantages.

a. The load ring will be expensive to produce and the load connectors and displacement sensor mounts will be critical for uniform results.

b. The displacement sensors must be very accurate as changes are small. This will probably require an LVDT with its inherent problems.

c. Strain gage sensors will be more difficult to mount and weatherproof than 2, 3 and 4 and the increased sensitivity will be of little advantage.

Due to the limits and difficulties with different load sensing transducers it will probably be desirable to utilize the unit that can be most easily made that will still give good sensitivity. It appears that units 2, 3 and 4 come closest to meeting this need. All three utilize strain gage sensors which will require determining an electronic method for zero calibration. Finding a way to unload the transducer for zero calibration will probably be impractical. If an electronic method for zero calibration cannot be determined then it will be necessary to go to units 1 or 6.

Section 6. Select the angle sensors for the angle of the support frame to Ts (β) and the angle of the outgoing line (θ).

The angle sensors are mounted on pin P between the load support frame and the load sensor. They can be connected to the frames or to the pin. The simplest mounting method would probably be to mount a sensor on each end of the fixed shaft then let the case rotate with the tension sensor or the line angle sensor. The (θ) sensor will need to operate over $\pm$ 15° and the zero point needs to be accurately determined. This could be done mechanically as the zero should not shift but it might be more desirable to determine the zero electronically. The electronics must be able to determine both positive and negative values and then determine the sine and cosine of the angle.

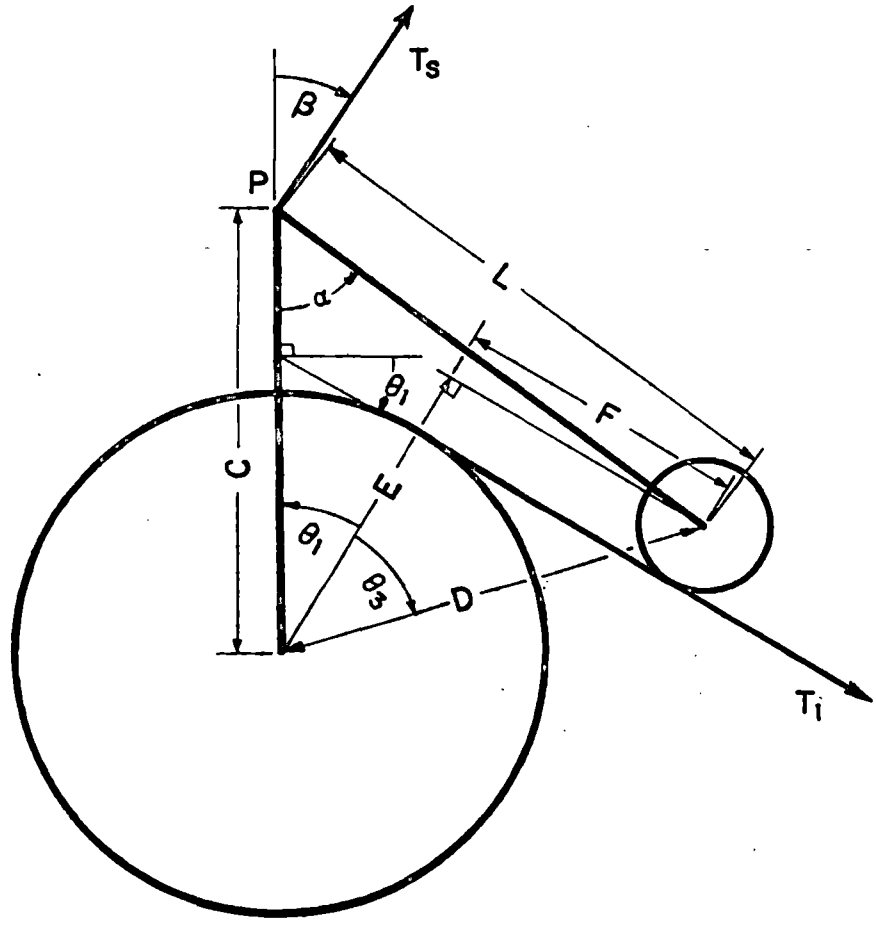


Figure 6.1 Angle Sensor for the Outgoing Line.

The (θ) sensor should be able to operate from zero to plus 90°. Most of the operation should be in the 10 to 30° range but a drive unit with 180° wrap could even exceed 90° when the torque is applied. The sensor must be able to sense both positive and negative values so that the same sensor can be used on drives with either clockwise or counterclockwise rotation of the sheave. Both a mechanical and electronic zero will be needed for accurate measurement. This angle must be converted into sine and cosine to determine the approach and exit angles of the cables.

The sheave that rests on the outgoing line should have a pitch diameter of 6 to 7 inches and have a guide similar to the load sheaves to pass the cable connectors. The mounting hub between the sheave and the support arm should be bolted in place so it can be used for either a left or right drive. A tension spring should be used to hold the follower sheave against the cable.

Section 7. Select the operating parameters for the cable speed sensor.

The cable speed will vary from zero to 500 feet per minute. This will give a sheave speed of zero to 140 RPM. A speed sensor is needed to indicate these speeds with a high degree of accuracy as the actual speed compared to the desired speed will be the primary control function. The Terrell HD1 - Style CB gear box has an enclosed low speed shaft and the hydraulic motor is also enclosed. It will be necessary to obtain an open cover for the end of the low speed shaft to use a shaft mounted sensor. An alternative would be to drill holes in the plate on the side of the sheave and use a magnetic pickup for rotation detection. A shaft mounted sensor may be more accurate and could be used with a Terrell HD1 - Style CP gear box. This would cause redesign of the sheave hub and shaft but no change in the operational parameters.

Section 8. Develop a hydraulic motor speed control plan.

This analysis compares different throttling valves for load and speed control. A standard throttling valve is a needle valve; however, a specially designed spool valve may be more desirable for control over a wide speed range.

The energy absorbed by a throttling valve can be expressed as:

$E = \Delta P \times V$

E = energy loss in inch pounds per minute

ΔP = The pressure drop through the valve in psi

V = The flow rate in cubic inches per minute

The energy absorbed can be changed to horsepower by dividing by 396,000.

(12 in/ft x 33,000 ft lb/(min HP) = 396,000 in lb/(min HP)

Best control will be obtained if the throttling valve gives linear changes in energy for linear changes in the valve. This would be uniform rotation for a needle valve or linear displacement of a throttling spool valve.

When the system is operating at a constant speed, which would be normal operation, the flow will be constant. The energy change will then depend on the position of the load between towers and the grade. The pressure drop P must be changed to absorb the change in load and a linear change in P would give a linear change in energy.

The orifice flow equation is:

$$V = C A \sqrt{\Delta P}$$

$$P = \left[\frac{V}{C A}\right]^2 = \frac{C1}{A^2}$$

C1 is a new constant that is a function of flow and the orifice constant. The value of C1 will change with different flow rates which are proportional to different cable speeds.

Four different throttling valves were evaluated to determine the linearity of control for each valve. These four valves were:

1. A conical needle valve
2. A round spool valve with a tapered rectangular orifice
3. A round spool valve with a variable depth rectangular orifice
4. A needle valve with a non-conical section

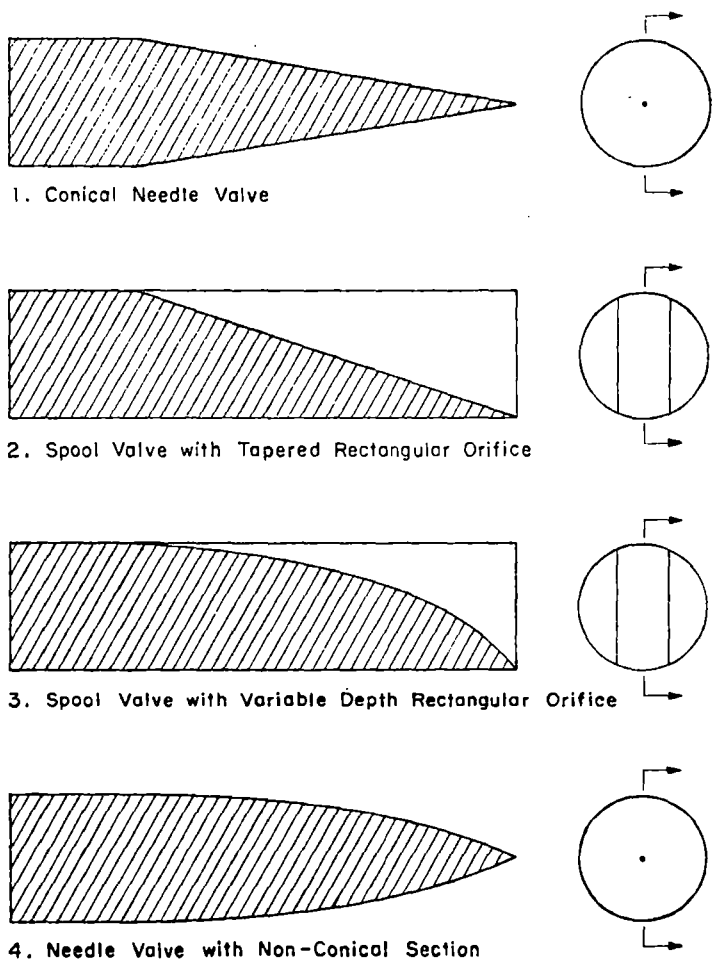

1. Conical Needle Valve
2. Spool Valve with Tapered Rectangular Orifice
3. Spool Valve with Variable Depth Rectangular Orifice
4. Needle Valve with Non-Conical Section

Figure 8.1 Cross Sections for Four Different Control Valves.

A cross section of these four valves is shown in Figure 8.1. The conical needle valve and the spool valve with the tapered rectangular orifice could be easily manufactured in a machine shop. The spool valve with a variable depth rectangular orifice and the needle valve with a non-conical section each must have the shape of the section defined by an equation. Construction would then require a numerically controlled milling machine or lathe for these two throttling valves. If this equipment is available they would be no more difficult to manufacture. The selection of the best throttling valve will depend on the linearity of response at constant cable speed. The equations for the valves were forced to have the same orifice area at two displacement points for comparison purposes. In this manner the energy changes between these two points can be compared. An area of 1 and 1/5 would give an area change of 5 to 1 and an energy change of 25 to 1. This is about the range of energy change expected in the hydraulic system as we go from a minimum pressure drop of 100 psi to a maximum pressure drop of 2500 psi. It is probably a greater range than needed as most control functions will probably be over a much shorter range. They should, however, all operate at the low energy range when there is no load on the system. The orifice area range for analysis was chosen as 1 and 1/4 to give an area change of 4 to 1 and an energy change of 16 to 1.

The equations for the four throttling valves are described as follows:

1. Conical Needle Valve

$d = X1 \times D$

d = the diameter of the needle valve at any point X1

D is the maximum diameter of the valve

X1 is measured from the point of the needle valve and is equal to 1 at the maximum diameter of the valve. The area at any point on the valve is

$$A = \frac{\pi}{4} (D^2 - d^2)$$

$$A = \frac{\pi}{4} (D^2 - (X1\ D)^2$$

$$A = \frac{\pi}{4} D^2\ (1 - X1^2)$$

Assuming that A = 1 at X1 = 0.2 then it is possible to solve for X1 at A = 0.25

$$A = 1 = \frac{\pi}{4} D^2\ (1 - 0.2^2) = \frac{\pi}{4} D^2 \times 0.96$$

Therefore $\frac{\pi D^2}{4} = 1/0.96$

at $A = 0.25 = \frac{\pi}{4} D^2\ (1 - X1^2)$

$$0.25 = (1 - X1^2)/0.96$$

$$X1^2 = 1 - 0.25 \times 0.96 = 0.76$$

$$X1 = 0.87178$$

The conical needle valve should, therefore, be checked for linearity from X1 = 0.87178 to X1 = 0.2

2. Spool Valve with Tapered Rectangular Orifice

$$y = D - X2\ D = D(1 - X2)$$

y is the depth of the orifice

D is the diameter of the orifice

X2 varies from zero to one with zero at y = D to 1 at y = 0

The area for an orifice with width w at any point on the valve is

$$A = w\ y = w\ D\ (1 - X2)$$

For an analysis similar to the conical needle valve the area A = 1 is defined at the point where X2 = 0.2

$$A = 1 = w\ D\ (1 - 0.2) = w\ D\ (0.8)$$

$$w\ D = 1/0.8 = 1.25$$

at $A = 0.25 = w\ D\ (1 - X2) = 1.25\ (1 - X2)$

$$X2 = 1 - \frac{0.25}{1.25} = 0.80$$

The spool valve with a tapered rectangular orifice should be checked for linearity from X2 = 0.2 to X2 = 0.8

3. Spool valve with variable depth rectangular orifice.

The equation for the depth of the orifice was chosen as $y = D\ C^{X3}$

D is the diameter of the spool valve

y is the depth of the orifice at any value of X3

X3 is measured from the end of the valve

The constant C was selected from a short linear portion of the tapered rectangular slot orifice. The value of C was checked from C = 0.5 to C = 0.9 and the value was not critical as the scale factor for the exponent X would compensate for changes in C when the overall dimension of the spool valve were determined.

The area of the orifice is:

$A = wy = D\ w\ C^{X3}$

w is the width of the slot

C is a constant selected at 0.75 for this analysis

A = 1 was defined at X3 = 1.0

$A = 1 = D\ w\ 0.75^{1}$

$$D\ w = \frac{1}{0.75} = 1.3333$$

at $A = 0.25 = D\ w\ .75^{X3}$

$0.75^{X3} = 0.25 \times 0.75 = 0.1875$

$X3 \ln 0.75 = \ln 0.1875$

X3 = 5.8188

The spool valve with a variable depth rectangular orifice should be checked for linearity from X3 = 1.0 to X3 = 5.8188

4. Needle Valve with Non-Conical Section

The diameter for the needle valve with a non-conical section was chosen for a diameter that changed in a fashion similar to the variable depth rectangular orifice. The diameter at any point X4 from the point of the valve is:

$$d = D - D\ 0.75^{X4} = D\ (1 - 0.75^{X4})$$

d is the diameter at any point X4

D is the maximum diameter of the needle valve

X4 is the distance from the point of the valve

The scale factor for X4 will need to be determined for the actual physical dimension of the valve desired.

$$A = \frac{\pi}{4}\ (D^2 - D^2(1 - 0.75^{X4})^2) = \frac{\pi}{4}\ (1-(1-0.75^{X4})^2)$$

A = 1 was defined at X4 = 1

$$A = 1 = \frac{\pi}{4} D^2 (1 - (1-0.75)^2) = \frac{\pi}{4} D^2\ (1 - 0.0625)$$

$$A = \frac{\pi}{4} D^2\ 0.9375$$

$$\frac{\pi}{4} D^2 = \frac{1}{0.9375} = 1.06667$$

$$\text{at } A = 0.25 = \frac{\pi}{4} D^2\ (1 - (1 - 0.75^{X4})^2)$$

$$\frac{0.25}{1.06667} = 1 - (1 - 0.75^{X4})^2$$

$$1 - 0.75^{X4} = \sqrt{1 - 0.25/1.06667}$$

$$0.75^{X4} = 1 - \sqrt{1 - 0.25/1.06667} = 0.1250$$

$X4 \ln 0.75 = \ln 0.125$

$X4 = 7.2283$

The non-conical needle valve should be checked for linearity from X4 = 1 to X4 = 7.2283

The equations for orifice area are.

1. Conical Needle Valve

$$A = \frac{1}{0.96}(1 - X1^2) \qquad \text{for } X1 = 0.2 \text{ to } 0.87178$$

2. Round Spool Valve with a Tapered Orifice

$$A = 1.25\,(1 - X2) \qquad \text{for } X2 = 0.2 \text{ to } 0.8$$

3. Round Spool Valve with a Variable Depth Orifice

$$A = \frac{1}{0.75}(0.75^{X3}) \qquad \text{for } X3 = 1.0 \text{ to } 5.8188$$

4. Needle Valve with a Non-conical Section

$$A = 1.06667\,(1 - (1 - 0.75^{X4})^2) \qquad \text{for } X4 = 1 \text{ to } 7.2283$$

The energy absorbed in the orifice is proportional to the reciprocal of the square of the area or (k is a constant for any valve design):

$$E = \frac{k}{A^2}$$

The energy absorbed by the throttling valves when operating at constant speed is shown in Figure 8.2. None of the valves give linear energy change over the entire range. The spool valve with the variable depth rectangular orifice (3) was the most linear and the conical needle valve (1) was the least. Therefore, the best shape for the throttling valve will be a spool valve with a variable depth orifice defined by the equation:

$$y = D\,(C^{X3})$$

The units for X3 must be redefined to fit the dimensions of the proposed valve. At X3 = 0 the orifice depth will be equal to D. The orifice depth will

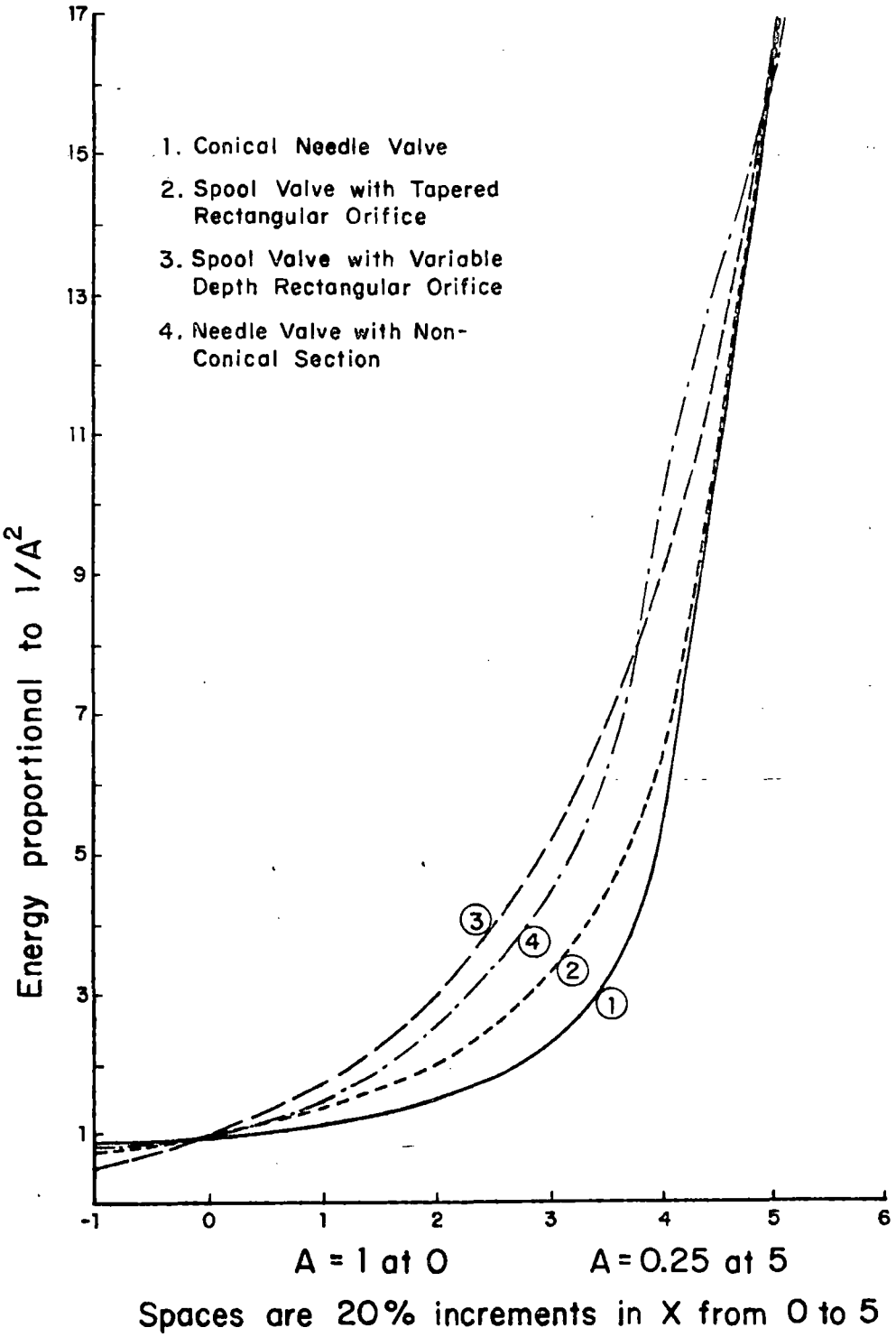


Figure 8.2 Energy Absorbed at Constant Speed by the Throttling Valves.

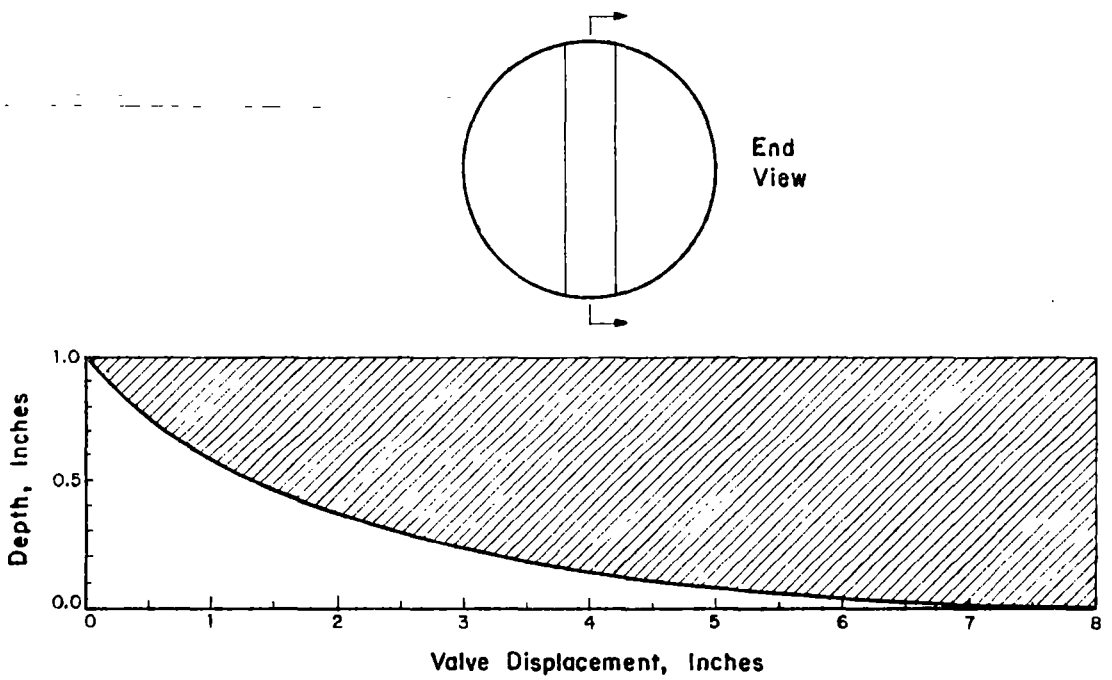


Figure 8.3 Cross Section of the Variable Depth Orifice Valve.

not approach zero until X3 approaches infinity. It is necessary, therefore, to select a minimum value for y. For this analysis the minimum value for y was assumed to be 0.020 inches and the diameter D was selected as 1.000 inches. It is also assumed that it is desirable to obtain this change in orifice depth over a spool valve motion of 8 inches.

X3 varies from X3 = 0 to the value of X3 when y = 0.020 inches.

Solving for X3 gives:

$y = 1.000\ (0.75^{X3}) = 0.020$

$0.75^{X3} = 0.020$

$X3 \ln 0.75 = \ln 0.020$

$X3 = 13.598425$

The value of X3 of 13.598 represents a physical dimension of 8 inches. Therefore, the units of X3 are 1.6998 units per inch or 0.588303 inches per unit of X3.

The cross section for this valve is shown in Figure 8.3.

These equations used to determine Figure 8.3 are the dimensions of the finished piece and are not the coordinates needed by the milling machine. It will probably be necessary to rewrite the equation in real dimensions then differentiate to get the slope at each point along the curve. This slope and the radius of the milling cutter would be used to determine the coordinates of the milling machine. The vertical coordinate would change by the radius times the cosine of the angle and the horizontal component would change by the radius times the sine of the angle.

The temperature gain due to a throttling process is the best way to describe the effect of energy absorption. The energy absorbed is equal to the pressure drop times the flow. The heat gain is proportional to the pressure drop and the specific heat of the fluid.

The properties of hydraulic fluids are:

ρ density is 0.031417 pounds per cubic inch

C_p specific heat is 0.511 BTU per pound per degree Fahrenheit

The mechanical equivalent of heat is 778 ft-lb/BTU

The energy produced by the throttling process is:

$E_p = \Delta P$ lb/in^2 x V in^3/min x 1 ft/12 in.

The energy gained by the fluid is:

E_p = V in^3/min x ρ lb/in^3 x ΔT^oF x C_p BTU/lb x 778 ft-lb/BTU

Note that the mass flow appears in both equations. It does determine the total amount of heat absorbed by the fluid but the temperature gain T is independent of the flow rate and is determined entirely by the pressure drop and constants for the fluid of concern. Solving for the temperature gain then yields.

$$\Delta T^oF = \frac{\Delta P}{12 \times 0.031417 \times 0.511 \times 778} = \frac{\Delta P}{150}$$

Therefore, the throttling process used for braking will result in a temperature gain of 1^oF for every 150 psi of pressure drop. The maximum temperature gain will be determined by the maximum pressure drop. At a pressure drop of 2500 psi the oil temperature will increase 16.7^oF each time it passes through the throttling valve. A safe oil temperature is 250^oF or less. If we use an oil cooler with an output oil temperature of 150^oF (100^oF temperature drop using atmospheric air as the cooling medium) then the oil could circulate through the throttling valve 100/16.7 = 15 times to reach the maximum temperature gain. The charge pump would, therefore, need a capacity of at least 1/15th of the pump capacity to provide the excess flow for cooling purposes.

Section 9. Design a hydraulic motor displacement control.

Variable volume motors and pumps will be needed. These motors and pumps are available with external lever controls on small units and with internal servo controls on large units. The units used in the test facility are small and those in the full scale system may still be small enough to utilize external lever controls. Most cam displacement controls operate over a range of ± 10 to 17 degrees. The displacement is proportional to the tangent of the cam angle as:

$$D = D_{max} \times \text{Tan}\phi / \text{Tan}\phi_{max}$$

The change in displacement will be proportional to the tanϕ and in this range the tangent is nearly linear.

The control lever that operates the cam can be any length desired but usually is at least 4 inches long. A linear actuator attached to the control lever will move the lever approximately as a function of the sine of the lever angle. Again this is a nearly linear portion of the sine curve and it is nearly the same as the tangent of the angle. Over a 10° range the difference would be only 1 1/2% and over 17° the difference is 4%. For small increments of change, as would be used for control purposes, there will be no noticeable difference between the sine input from the linear actuator and the tangent output of the cam.

The total motion required for the linear actuator will be.

$$M = 2\ L\ \sin\phi_{max}$$

M is the motion of the actuator in inches

L is the length of control lever arm

ϕ_{max} is the maximum angular change of the cam control from the neutral or zero displacement position.

The multiplier 2 is used for a control that must operate from full forward to full reverse and would be the longest linear change that the system could

utilize.

For a 4 to 10 inch lever arm with a 17° cam the linear actuator would need to have a stroke of:

$M = 2 \times (4 \text{ to } 10) \times \text{Sin } 17^{\circ} = (2.34 \text{ to } 5.85)$ inches

Most of the control levers will not be operated over the entire range and will probably be 1/2 of this amount or less.

The linear actuators should be 12 volt DC devices such as convertible top controls. These are available from companies such as Borg Warner with strokes well within the range needed.

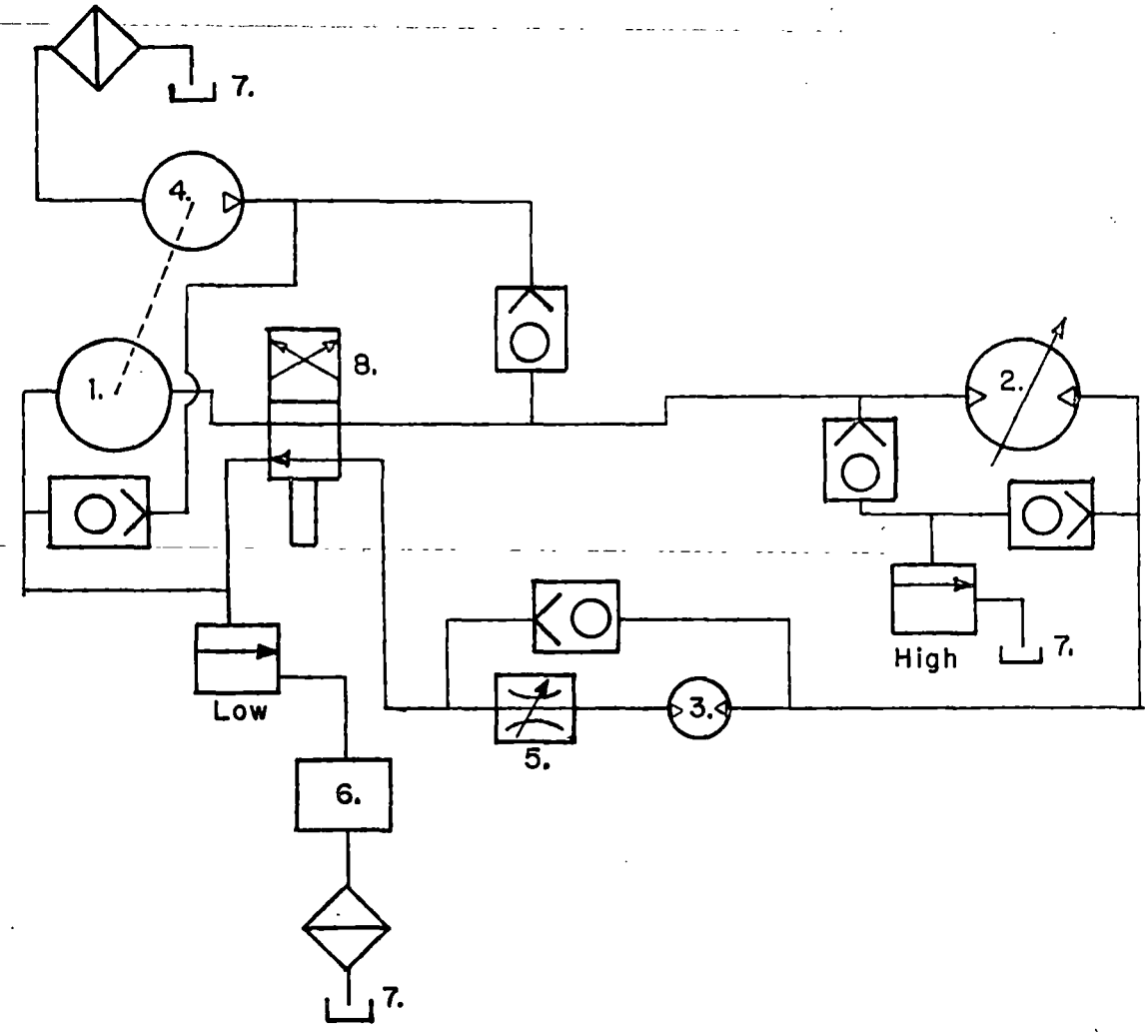
It may be desirable to attach an angle position indicator to the control lever so that the angle can be used as a speed indicator and the rate of change of angle can be used as a feedback signal for stability of the control function.

Section 10. Determine the location and connections for the necessary pressure relief valves.

Pressure relief valves are required to prevent failure of the hydraulic components due to unforeseen overloads or the system operating in an unforeseen or uncontrolled mode. (Such as a log causing a motor to rotate backwards during an emergency shutdown.)

The hydraulic system shown in Figure 10.1 consists of a main fixed displacement pump, a variable displacement motor, a secondary fixed displacement motor to drive the alternator, a charge pump, a throttling valve, a heat exchanger and a reservoir. The main lines should all be relieved at approximately 3000 psi and the charge lines are relieved at approximately 50 psi. A direction flow control valve will also be needed to allow both lines to feed out in the same direction while inserting or removing cable segments.

There are three main portions of the plumbing system. The input to the main pump between the throttling valve and the pump, the line from the main pump to the variable displacement motor and the line from the variable displacement motor to the throttling valve. All three of these areas must have the potential to receive low pressure charge flow to prevent cavitation, if one motor turns faster than design, plus high pressure relief if something stops. In addition, a check valve is needed to bypass the throttling valve in the event the main fixed displacement motor would rotate backwards. This check valve also supplies charge pressure between the valve and motor if needed. Note that the alternator drive is also bypassed. This will prevent backwards rotation of this unit.



1. Main fixed displacement pump
2. Variable displacement motor
3. Secondary fixed displacement motor
4. Charge pump
5. Throttling valve
6. Heat exchanger
7. Reservoir (1 only)
8. Direction control valve

Check valves

Pressure relief valves

Figure 10.1 Hydraulic Circuit.

Section 11. Determine the charge pump capacity needed for anti-cavitation control and circulation to the heat exchanger.

The charge pump maintains a low positive pressure (approximately 50 psi) on all portions of the hydraulic system that might normally or due to unforeseen circumstances create a vacuum. Under normal operation most or all of the charge flow will flow into the intake of the main pump. This inflow will compensate for any leakage from the pumps and motors and the excess will force hot oil coming from the throttling valve through the low pressure relief and the heat exchanger. The charge fluid comes from the reservoir and should be as much as 100°F cooler than the circulating oil. The charge pump capacity is proportional to the main pump capacity. Both run from the same shaft at the same speed, therefore, capacities are proportional to displacement.

The charge pump needs about 5% of the pump flow to prevent cavitation and to compensate for pump and motor leakage. An additional 1/15th or 7% is needed to circulate cooling oil. The total displacement required for the charge pump is, therefore, 12 to 15% of the main pump displacement. Larger displacements would not be harmful and would increase the cooling flow, therefore, reducing the temperature drop needed in the heat exchanger. At 15% of pump displacement and 50 psi pressure drop the energy required for the charge pump is only 0.3% of the potential energy in the main pump.

Section 12. Determining the Filter and Reservoir requirements.

The filter requirements for a hydrostatic drive are very critical for long life of the precision components. A 10 micron replaceable filter should be placed on the intake to the charge pump to prevent any dirt or metal particles from entering the system. This filter should have the bypass blocked. In this system a filter blocked indicator may be needed. This would be a vacuum gage on the suction side of the pump. A microswitch on a spring loaded piston would be a simple and reliable detector.

A second filter should be placed between the charge pump pressure relief and the heat exchanger. This filter is used to remove any wear particles that might be circulating in the hydraulic fluid. Pump pressure will move the fluid through the valve and a dirty filter bypass must be provided to prevent filter failure and oil loss.

The oil reservoir on hydrostatic drives can be relatively small as the majority of the oil is circulating in the system and the only oil circulating through the reservoir is equal to the charge pump flow. In addition the only volume change in the reservoir is due to heating and cooling of the oil as there are no volume changes due to piston extension and retraction. The oil volume circulating through the reservoir will be about 15% of the main pump volume. The main pump was selected with a displacement of approximately 1.643 in^3 and it will operate at a maximum speed of 730 RPM. This gives a flow rate of 1,197 in^3/min or 5.18 gallons per minute. Only 15% of this flow will go through the reservoir; therefore, the flow will be 180 in^3/min or 0.78 gallons per minute. The reservoir is sized to have a hold time in the reservoir to allow separation of dirt and entrained air. The reservoir also serves as a heat exchanger and is often made with large flat surfaces to increase heat transfer. A 3 x 20 x 20 reservoir is a fairly common size and it has a capacity of 5 gallons. At least 4 gallons of this capacity would be usable capacity with the additional gallon

for makeup and expansion. With 4 gallons the hold time in the reservoir would be 5 minutes. This is 2 1/2 times greater than the normal design of 2 minutes but will probably be justified as it will reduce the need for an additional heat exchanger.

The tank must include a breather filter in the filler cap to allow volumetric changes in the tank. A rust resistant internal coating is also useful to reduce contamination. The suction line should be near the bottom of the tank, but should have a lip above the bottom of the tank to permit some dirt to accumulate on the bottom of the tank without being sucked into the suction line. The return line to the tank should be below the fluid level, but above the bottom of the tank to reduce air entrainment and so it will not circulate dirt on the bottom of the tank.

Section 13. Determine the heat exchanger requirements for the system.

The heat exchanger should be able to dissipate all the heat generated in the throttling process. This will amount to as much as 100°F temperature drop with a flow rate of 180 in^3/minute or 0.78 gallons per minute. The reservoir will probably be able to handle 12° to 15° of this difference, therefore, the heat exchanger must be able to achieve an 85° temperature drop when the incoming fluid is at 250° with a flow rate of 0.78 gallons per minute. This temperature drop should be fairly easy to achieve as the design outlet temperature of 165°F is still well above any ambient air temperature that would be expected. There is a good chance that this cooling capacity can be obtained without a fan. A fan could easily be provided either as a belt drive from the alternator shaft or as a 12 Volt DC automotive fan with thermostat temperature control.

The specifications to be sent to a heat exchanger manufacturer should include the following:

Heat exchanger specifications

Flow 0.78 gallons per minute

Inlet temperature Up to 250°F

Outlet temperature 165°F or lower

Ambient air temperature -10° to 100 °F

Pressure atmospheric to 50 psi

Air flow natural circulation or fan

Section 14. Determine the alternator, regulator, battery and alternator drive needs.

The electrical components should be of automotive grade to be readily available at a minimum cost. A 10 to 20 amp alternator should be adequate unless a large fan is needed for cooling. It may be more desirable to go as high as 40 amp if this is a more standard readily available size. Excess capacity will not be harmful and the decision should be made on the basis of availability and cost. The regulator should be of the solid-state type and one supplied by the manufacturer of the alternator.

The battery size will be determined by the needs of the cooling fan. The other electrical requirements will be relatively low. A small garden tractor battery of 60 to 90 ampere hours capacity will probably be adequate.

The drive to the alternator should be a small high speed hydraulic motor. The best drive arrangement would be a direct shaft connection. The oil flow through the motor will be as much as 1,197 in^3/minute. A motor with a displacement of 0.33 in^3/revolutions would operate at 3600 RPM under these conditions. If the design speed for the alternator is 2000 RPM the motor displacement would be 0.6 cubic inches. It will probably be possible to find a motor within this size range and an alternator that will develop 50 % to full capacity at a speed of 1000 RPM or more.

Section 15. Determine the pump and engine needs for hydraulic power supply for typical load units and the main drive unit.

The design maximum cable power at any sheave is 7.6 HP. The input power to provide this energy would be.

HP = 7.6 / (Em x Ep x Ee)

Em = Hydraulic Motor Efficiency (0.75 to 0.95) use 0.8

Ep = Hydraulic Pump Efficiency (0.75 to 0.95) use 0.8

Ee = Efficiency of the power unit use 1.0 for an electric motor,0.85 for a diesel engine and 0.75 for a gasoline engine

The power units would, therefore, require a 12 HP electric motor, a 14 HP diesel engine or a 16 HP gasoline engine.

The main drive unit could be larger as 180° wrap would permit a pull of 0.613 times the tight side tension at c = 0.3. At a maximum tension of 5000 pounds, the pull could be 3065 pounds. At 500 feet per minute this would produce cable power of 46 HP. This would be impractical as the slack side tension would be only 1935 pounds. Half of this power, however, would be very practical, therefore, the primary drive unit might be designed for maximum pull of 1500 pounds. This will develop a cable power of 22.7 HP. The input power units will need to be 35 HP for an electric motor, 42 HP for a diesel engine and 47 HP for a gasoline engine. This power level could still be applied through the Terrell HD1 gear box.

Section 16. Design a pump displacement control for loading devices (Pressure Control) and the drive unit (Speed Control).

Test devices that apply power to the cable must have variable speeds that are determined by the displacement of the supply pump. The control lever is similar to that on the variable displacement motor, and control can be achieved with the same types of linear displacement device. The control, however, must operate over the entire forward speed range of the pump and even a low speed reverse motion may be desirable for some test conditions.

The load units will be variable speed devices that simulate loads by applying force (either pull or braking) to the line. The pressure at the pump outlet will be proportional to the load. The pump should adjust displacement to maintain a predetermined pressure or pull on the line. The control signal will be the difference between a computer determined pressure to simulate the load and the actual pressure measured at the pump outlet. A feedback signal proportional to the rate of pressure change should be used to provide system stability.

The main drive unit will utilize speed controls similar to the speed control on the braking units. The motor displacement will be adjusted rather than adjusting a throttling valve.

Section 17. Develop the loading device and load simulation method for the device.

The loading devices will use the same sheave, sheave guides, gear box and fixed displacement hydraulic motors as used on the braking devices but with an external power supply and variable displacement pump that will deliver the flow needed for the motor. The sheave pattern required for the loading units is shown in Figure 17.1.

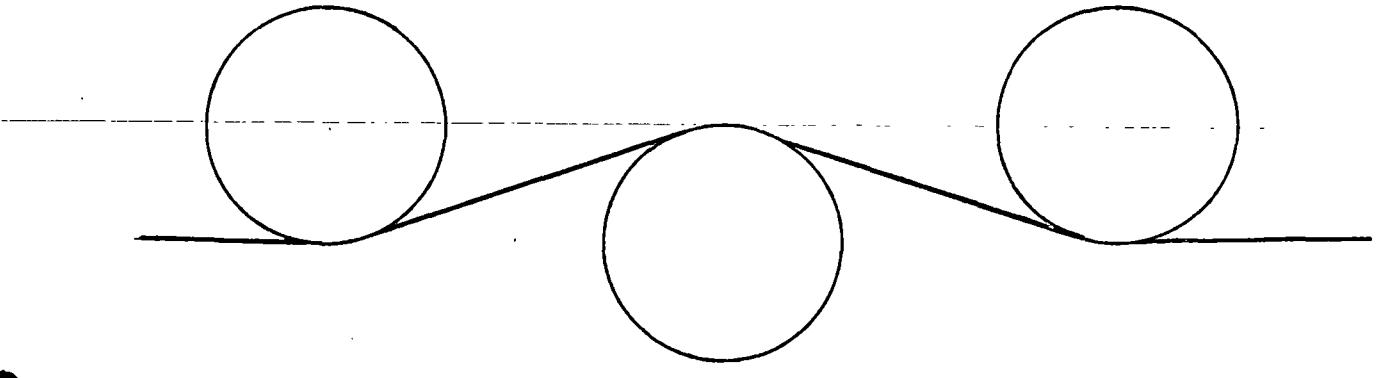


Figure 17.1. Sheave Pattern for the Loading Unit.

The sheaves are placed in a pattern to permit the cable connectors to pass readily between the sheaves. The wrap angle is not critical, but must be 20 degrees or more. Placing the incoming and outgoing cables in line with the center of the drive sheave will simplify alignment of the drive unit with the line. The drive unit will be supported in a horizontal plane over the cable as shown in Figure 17.1. It will be held in place by cables to the left and right along the line and by a vertical support to carry the drive weight. The vertical support should be counterbalanced so that it can move in a vertical plane when a load passes through the drive. The two idler sheaves could be smaller than the drive sheave. This will increase fatigue wear on the cable, but this should not be a problem in the test facility.

A desired load will be predetermined to simulate the load change of a log moving between two supports. This will be a load shown as a pressure that will change as a function of time. The sequence of load changes as a function of time will be the primary portion of the system test sequence.

Section 18. Develop the end sheave plan including the mounting and cable take-up.

The end sheave will be an open-sided sheave with a mounting frame similar to the tower idler units. It will be connected by a 5+ ton winch for cable movement and take-up. A cable tension sensor (probably a hydraulic load cell with pressure gage) will probably be needed to prevent overload when moving and adjusting the end sheave. This overload protection could be obtained with a pressure relief on a hydraulic motor powered winch if the winch drum diameter remains nearly constant. This can probably be done by a winch with a drum wrap of only one or two layers. More wraps could be used with large diameter drums, however, this would increase the weight and cost of the winch. The winch with drive unit and skid frame can move itself into the desired position.

Section 19. Develop an idler sheave plan including mounting.

The idler sheave will have the same sheave and cable connector guide as the drive sheaves. The sheave will be fitted with a bronze bearing requiring only occasional lubrication. The solid hub for the bearing will attach to a 2 x 4 tube. The attachment point will be a clevis pin identical to the tower braking units. Figure 19.1 shows the plan for the idler sheaves.

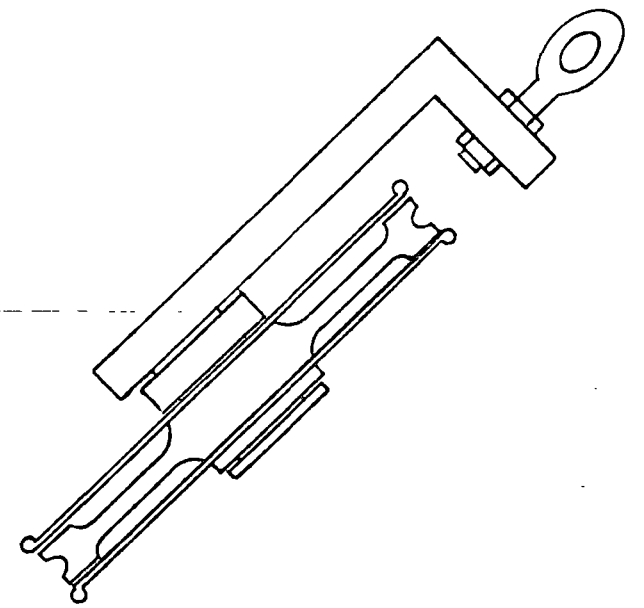


Figure 19.1 Idler Sheave with Bronze Bushing

A star wheel retainer will be attached to the axle of the idler sheave, and the drve axle of each braking sheave, to hold the cable and load as the cable attachment link and the load pass through the sheave. This star wheel is shown in Figure 19.2 on the next page. The star wheel will be mounted on a self aligning roller bearing to permit the star wheel to freewheel relative to the sheave. It will, however, flex against the sheave and be driven by friction when a load is applied.

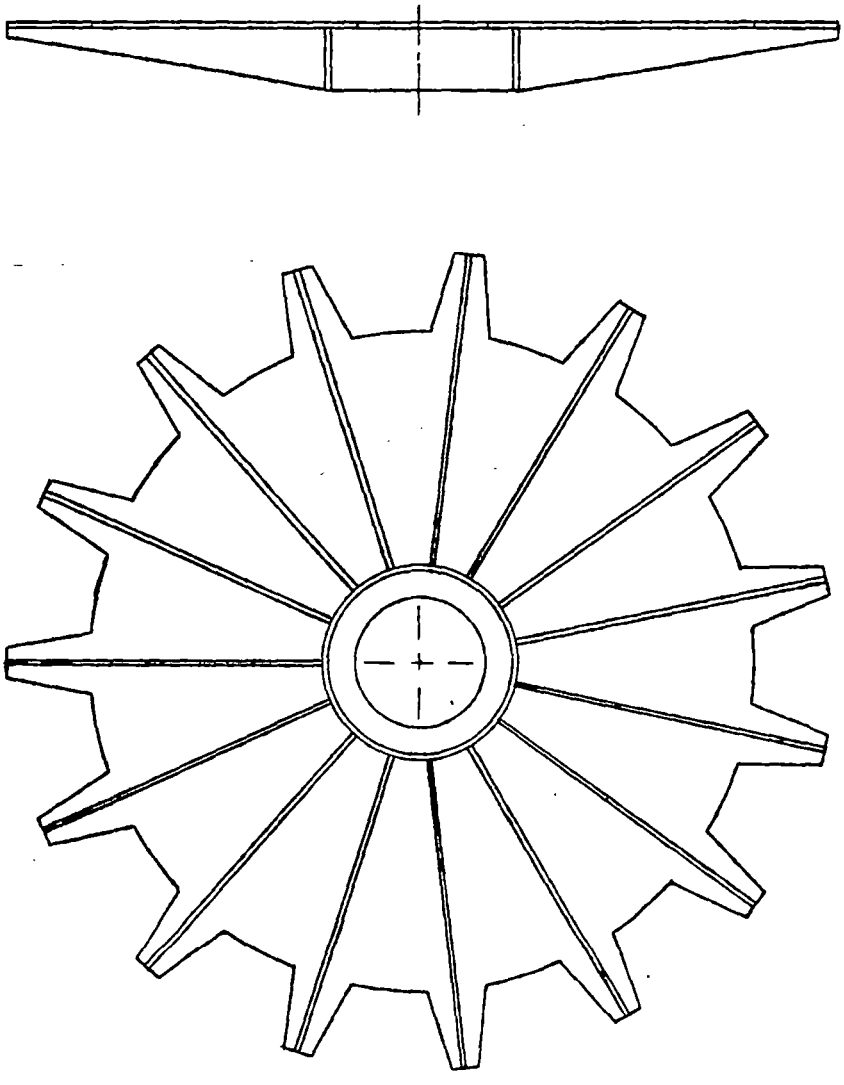


Figure 19.2 Star Wheel Retainer to Pass Connector Link and Choker

Section 20. Design the main drive unit including sheave or sheaves, motor, pump with speed and torque control.

The main drive unit will have a wrap angle of 180° on a sheave with sheave guides identical to the tower braking sheaves. The increased wrap angle permits tension differences up to 60% of the tight side tension when c = 0.3. This is probably excessive as it leaves the slack side tension near 40% or 2000 pounds or less. A more logical tension difference is about half of this amount or 1500 pounds. This is three times the power of the braking sheave which should be satisfactory at this point in the line. The cable power under this condition would be:

500 X 1500 / 33000 = 22.7 HP

An electric motor drive for this unit will require:

22.7 / (0.8 X 0.8) = 35 HP

The same gear box can be used as that on the braking units, but the pump and motors will be three times larger.

The main drive unit must be able to control speed and torque to prevent cable overload. It will require the same speed and tension measuring devices as on the braking units. The increased torque will not cause problems with the torque sensing device. The torque formula is:

$$T\,R = C\,Ts\,\cos\beta$$

$$T = \frac{C}{R}\,Ts\,\cos\beta$$

The angle β will not be excessive even though ΔT may increase 3 times. The increased wrap angle to 180° will cause Ts to increase faster than ΔT, therefore, $\cos\beta$ will not increase.

Section 21. Design the log loading unit including choker design, load support platform, cable positioning technique, load positioning technique and choker attachment device.

The loading unit in the woods must be capable of negotiating severe terrain to position the unit under the cable. The logs can be moved to various points along the line, but the loading points should be quite unlimited to reduce the amount of log moving. The undercarriage must be durable and able to operate over severe rocks and stumps. Sharp steering angles, good traction through tires and chains or tracks, self-leveling for steep slopes and grades plus small dimensions will all be required for maneuverability. Once the unit is in place it must be very stable to act as a work place for handling and attaching logs.

It is envisioned that the unit will have dimension of 8' x 16' or longer when in its mobile operating condition. It will have four tracks or wheels, all driven, that can be steered through very short angles. It will have winches on each corner to assist in the most extreme conditions. The wheels will be self-leveling for maximum traction on side slopes. An on-board operator will have complete control of all drive, steering and winching functions through hydrostatic controls.

Once in position the unit must be able to transform itself from a mobile unit to a stationary working platform. Four to eight outrigger feet would extend the dimensions of the unit to 24' x 28' or longer. (Eight foot outriggers with spike tooth pads to penetrate the soil surface). The outriggers are intended to form a wide stable platform and are not used to level the platform. They will be capable of lifting the unit from its wheels to give maximum weight on the outrigger pads. The winches could also be used when needed.

There will be a loading unit on each end of the platform to permit attachment of two chokers for full log support. The loading units are tower like

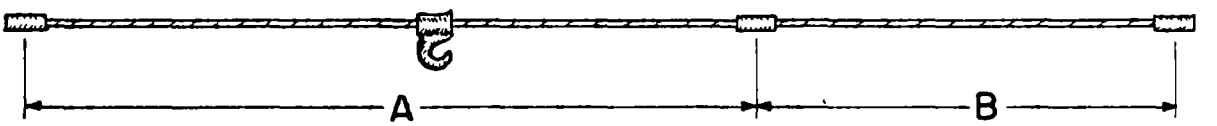
structures mounted on the 8 x 16 or longer frame. They can be leveled fore and aft as well as side to side so that the tower remains vertical even though the support frame is on extreme grades. The tower structure contains an enclosed cab or cage for one or two operators, cable positioning sheaves for loading and unloading, a log handling grapple and choker connecting and disconnecting devices.

The anticipated operating procedure would be as follows:

1. Move the unit to the desired position under the cable forwarding lines.
2. Stabilize the platform by rotating the outrigger legs onto the ground. The legs will be locked into place by hydraulic locking valves to allow them to adjust to variable terrain.
3. The operating tower is leveled then rotated around the center mast to position the loading sheaves under the incoming line and the unloading sheaves under the outgoing line.
4. The grapple fitted with a sheave is used to pull the lines down into the cable positioning sheaves.
5. The grapple is then used to pull a log from the deck and to place one end of it on a load attaching support.
6. The load attaching support is raised to bring the log up under the line.
7. The operator attaches a choker around the log and the other end around the load connecting device that encloses the cable.
8. The load attachment support is lowered to suspend the log from the choker and to set the load connecting devices.
9. The cable connector moving through the load connecting devices triggers its release to allow the choker to fall into the groove on the cable connector.

10. Loads being delivered to the woods, such as bundles of chokers, will be delivered to the unloading device where they will fall unto a platform or onto the ground.

The choker is a key element in the attachment process. It determines the position of the log under the cable, the amount of lift needed in the positioning platform and the design of the attachment and detachment devices. The choker used in the woods will use 3/8 inch wire rope and the test facility could use smaller wire rope. A 3/16 inch wire rope would be an exact scale, however, this is not important for the proof of concept analysis. The size wire rope used in the test facility will be determined by the availability of attachments such as swaged end fittings and sliding choker hooks. A choker identical to that used in the woods would not affect the test results even though it is heavier than needed. A typical choker with three swaged fittings and a sliding choker hook is shown in Figure 21.1.



A = *Length to connect tree, 30, 45, & 60 inches*

B = *Length to connect cable, 30 inches*

Figure 21.1. Typical Choker with Swaged Fittings.

The three lengths for dimension A would accommodate loads as follows:

60 inch Trees 18 inch and under

45 inch Trees 13 inch and under

30 inch Trees 8 inch and under

It may be possible to narrow this down to two sizes perhaps 60 and 40 inch.

In actual use choker portion B will be attached around the load connecting device. A device to hold the end fitting against the middle fitting will be described later. The log is then raised into position and choker portion A is attached around the log. Lowering the weight of the log onto the choker sets the release mechanism so it can be triggered by the cable connector.

The device to hold the two swaged fittings will consist of two forged or cast halves that completely enclose the two swaged fittings. A lever controlled cam will hold the two halves together. They can then be released by rotating the cam. The hinged end of the castings will be attached to the center swaged fitting. The proposed forged or cast fitting is shown in Figure 21.2.

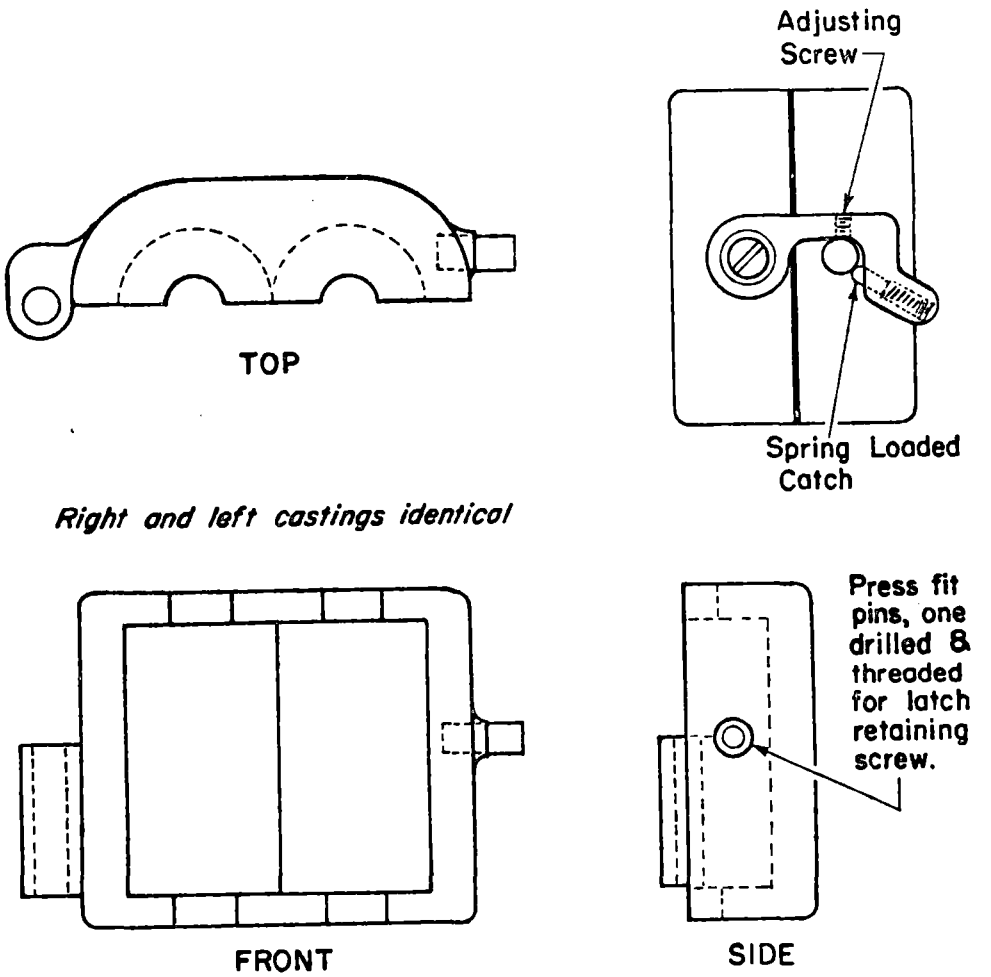
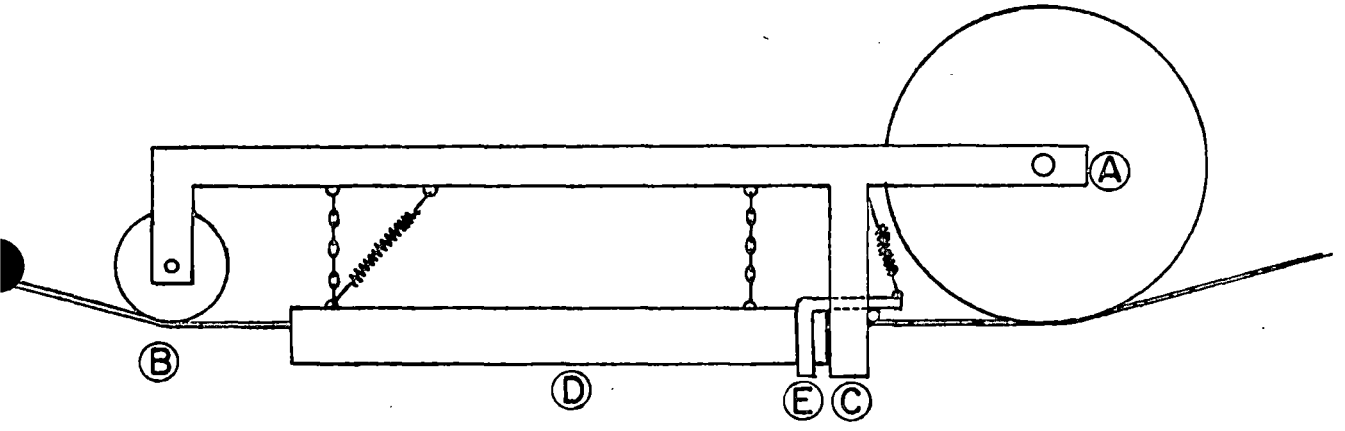


Figure 21.2. Casting to Hold Two Swaged Fittings in Place.

The locking cam goes slightly over center to lock the two halves in place. It is held in place by an off-center weight or a torsion spring. It is released by lifting the control lever.

The load attaching device is supported on the cable by two yokes that fit over the spindle of the sheave and a separate small sheave that rides on the cable. The shield over the cable is slightly larger than the cable connector. The shield consists of two sections, a fixed section near the sheave that includes the cable wiper and a spring loaded hinged section. The relationship between these components is shown in Figure 21.3.



A. Yoke on main sheave to support the load attachment device.

B. Cable alignment sheave.

C. Fixed cable shield rigid to frame.

D. Movable cable shield on a parallel linkage spring loaded to hold it against C.

E. Cable wiper to pull the cable off the end of D when D moves with the cable connector.

Figure 21.3. Choker Attaching Device.

The hinged section D is made in two parts as shown by the end view in Figure 21.4.

Figure 21.4. End View of Hinged Section.

In the closed position the cleats will contact the cable connector. This will cause the hinged section to move with the cable. The cable wiper then pulls the choker off the end of the cable cover and drops the choker into the groove on the cable connector. As soon as the cable and wiper are free of the hinged section it springs open to release the cable connector.

The choker is placed around the open hinged cable cover. When the load is lowered the weight on the choker closes the cover and sets the cleats and the cable wiper.

The load positioning platform is a hydraulic lift that clamps the end of the log against the front of the operator cage then raises the log into its working position. When lowered it moves below the frame to be clear of the log. The load positioning platform must be able to rotate to either side so logs can be loaded from either side of the machine.

The operator cage should be high enough to permit the operator to stand comfortably while working. The top, however, should be low enough to permit reaching above the cab to attach the choker to the choker attaching device. A working slot on each side of the front of the cage will be provided for attaching the choker to the log. The operator will then step into the back of

the cage to lower the load positioning device and set the attaching device. The operator's cage should rotate with the load positioning platform for proper alignment of the work area as shown in Figure 21.5.

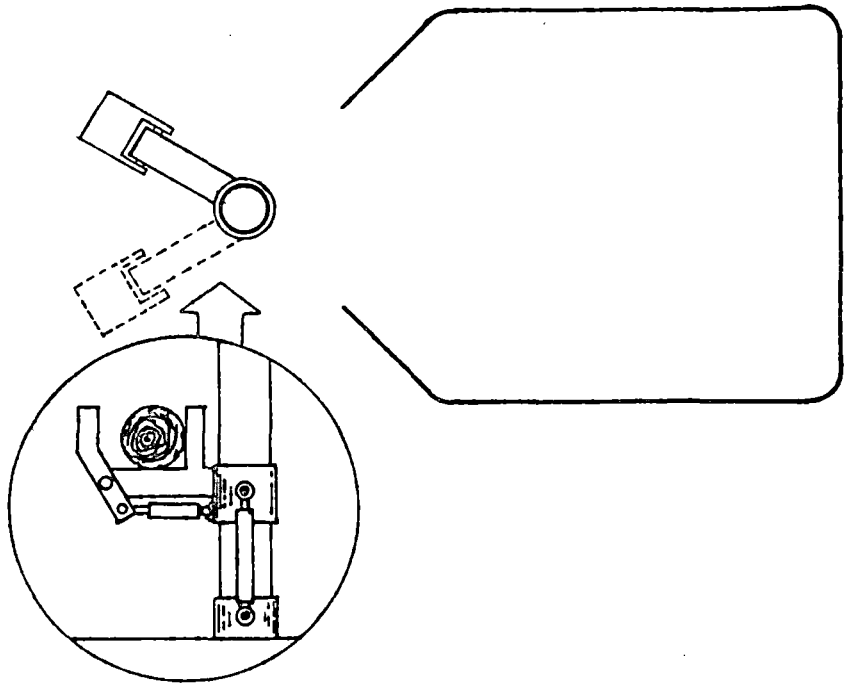


Figure 21.5. Operators Cage and Load Positioning Platform.

A similar loading device without the leveling platform or rotating cage will be needed at the landing.

The test facility will need everything except the mobile platform and the log handling grapple.

Section 22. Design the load unloading device including the choker disconnect technique and log handling device.

The log unloading device and a similar unit in the woods will attach to any idler sheave and is supported from the sheave spindle and a small sheave as shown for the attachment device. A V guide at the sheave catches the choker below the choker retainer casting. An inclined ramp moves closer to the cable andlifts the cam release lever. The ramp is spring loaded so it can hinge downward to release the choker as shown in Figure 22.1.

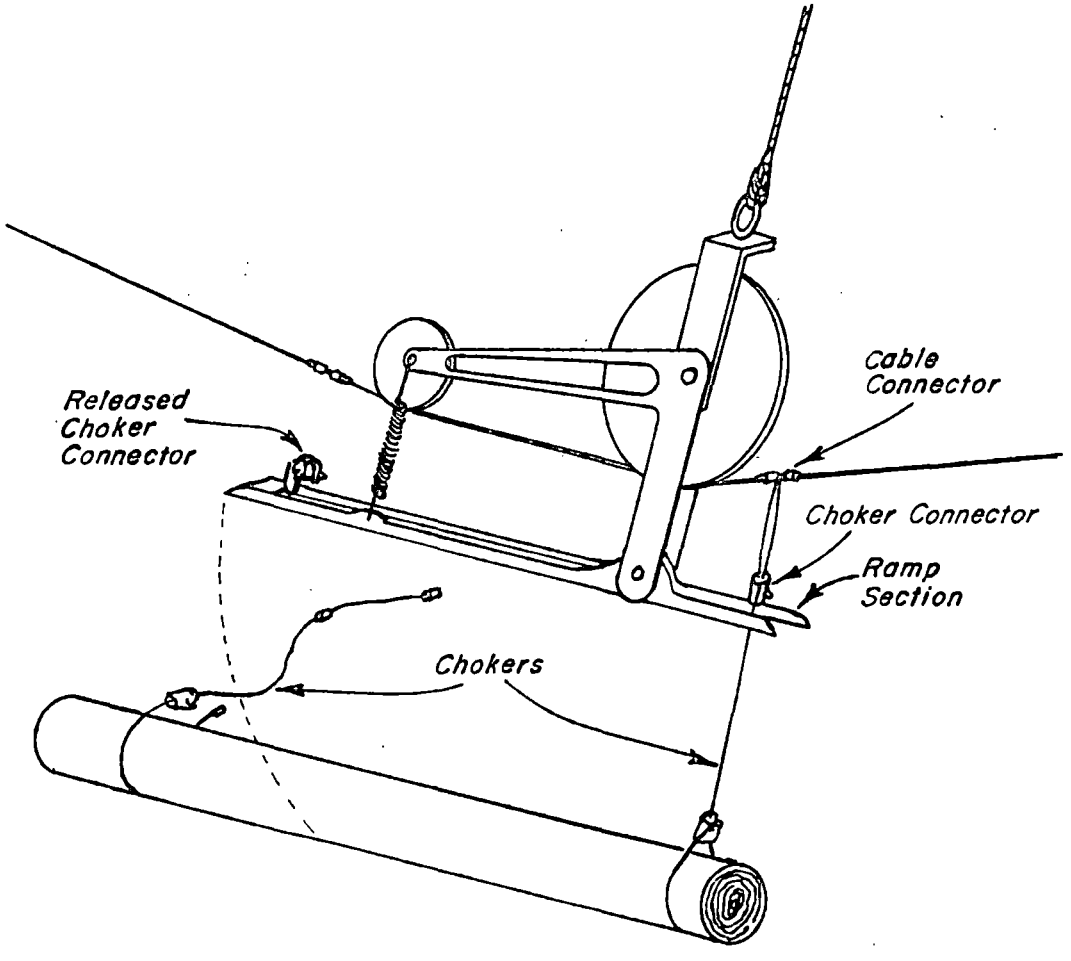


Figure 22.1 Log Unloading Device.

The log unloading device can be located at any idler sheave. At the landing the log will be released just after it has passed a support tower. A skid chute will be placed under the log to guide it into the proper position at the landing. The tower can, therefore, act as a side for a log accumulation

area. A similar device can be located in the woods either on towers or on the loading device.

Section 23. Design a cable tension control device that will compensate for changes in cable length during loading.

The cable tensioning device must adjust cable length for load changes on a continuous basis. It will be located at the landing and can be located either before or after the main drive sheave. A tension control device should be on the slack side of the drive. On most drives this will be on the outgoing line. The tension should be changed to maintain the proper tension difference across the drive. Under pulling conditions the tension on the outgoing line should be maintained at the design tight side tension less the maximum drive pull. For a 5000 pound maximum tension and 1500 pounds pull the design tension control should be 3500 pounds or less. This should be programmable so tension can be changed when needed and should approach the design maximum tension when braking.

The same tension measuring device used on the main braking sheave drive can be used to determine the tension on the tension control. An alternate method would be to utilize a tension sensor on an idler sheave. This may be more desirable as it separates this function from the drive control. No angle sensing is needed as the wrap can be maintained at 180° with zero torque. The force sensed will be twice the cable tension. A hydraulic cylinder might be adequate for a load cell with a pressure transducer, however, cylinder friction will give hysteresis that would not be observed with a strain gage load cell.

The tension control can be included with the cable segment insert and removal device. Due to the length of cable changed at one time an idler sheave on a winch will be a good cable length adjustment device. A hydraulic motor driven winch will, therefore, be used to adjust cable length as determined by the cable tension measuring device. The control signal will be the difference between the desired tension and the measured tension. The feedback signal for

stability could be the rate of winch rotation, but an easier signal to measure and one more nearly related to the control function is the rate of change in cable tension.

Section 24. Design a device and operating method for adding or removing cable segments.

The cable segments used in the woods will be 500 feet long. A winch moving an idler will, therefore, need to move 250 feet. Additional motion is needed to provide cable take up plus slack for loosening the cable prior to insertion of additional lines. A total length of 600 to 700 feet may well be needed. A single open-sided sheave with a movement of 350 feet would be the simplest for attaching and removing cable. If landing dimensions will not permit the 350 foot length then two extra idler sheaves can be added. This will reduce the distance to 175 feet, but double the winch force. Three sheaves at the winch will cut the distance to 116 feet and triple the winch force and 4 sheaves at the winch will decrease the distance to 88 feet. The most desirable condition will be the single sheave as the cable can be inserted by looping it over an open-sided sheave. If shorter lengths are needed it will be necessary to feed the cable through the sheaves.

A method for holding the cable ends while the cable is disconnected and a segment added or removed is needed. The cable tension should be slackened then held in a clamp so that the line will not feed out as additional slack is provided. The clamp should be made from two steel plates for each cable with a groove to fit tightly on the cable. The plates should be clamped with a hydraulic cylinder. Include an accumulator in the hydraulic circuit to maintain clamping pressure on the cables.

To remove a cable segment the line is rotated until the middle of a cable segment is near the idler sheave on the winch. The line is then stopped and cable tension is slackened somewhat for ease of handling. The cable ends are then clamped to hold the line while further slack is provided. The cable segment is then removed and the shortened line is then reconnected. The

reconnected line is slipped over the winch idler sheave, the tension tightened and the clamps released. Additional segments can be removed in a similar manner. The addition of line segments is a reversal of the above process.

Section 25. Design a cable support frame with attachments for positioning the sheave support frames.

The frame of the steel building used for the test facility might be adequate for supporting the cables, however the loads due to the cables are concentrated and not the normal design loads used for buildings. A more desirable system is to develop an internal steel frame that can withstand the cable tensions. The building frame may be able to withstand the tipping forces on the support frame. The vertical forces on the supports will be low compared to the horizontal forces due to cable tension (5,000 pounds vs 500 pounds).

The vertical support columns will have a length of 10 to 12 feet to give maximum height. The worst loading condition would be beam loading at mid-span and the force could be as high as 10,000 to 14,000 pounds. A pipe frame would be most desirable as the loads may be applied from different directions. The horizontal braces will be primarily under column loading. The loads could be split between top and bottom horizontal braces but it may be desirable to be able to place the loads at any height. The horizontal braces will therefore need to be able to withstand column loads of 10,000 to 14,000 pounds. Column lengths will vary from 10 feet to as much as 54 feet.

It will be difficult to anchor the support columns to the floor. The best system will be to construct a rigid frame approximately 10 feet from the floor. The cables will be connected just below this frame so that the forces are concentrated in this frame. Moving the cables nearer to the floor would require an additional rigid frame near the floor. It would be better to keep the floor clear and operating the cables near the 10-foot frame will not detract from the tests.

The space available for the test site is 40 x 40 feet. The ceiling height is 12 feet in most of the room. The heaters and the door rails decrease the headroom to 10 feet in these locations. A structural frame will be located in a

U shape in the west end of the room as shown in Figure 25.1. The frame will suspend the test equipment 10 feet above the floor. All cables will be kept close to the support frame so that loads will be carried by the truss frame shown in Figure 25.1. The system layout under the structural frame is shown in Figure 25.2.

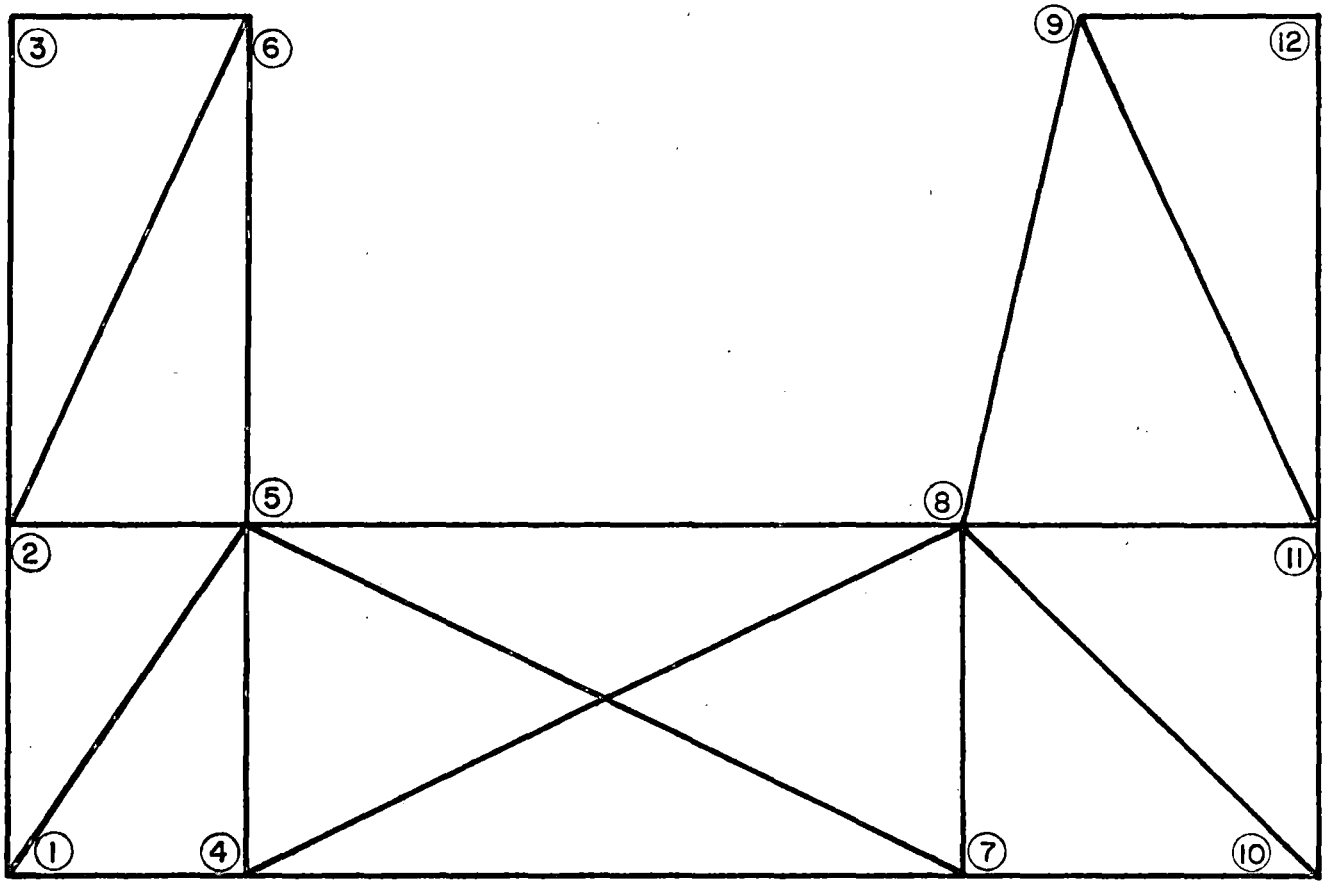


Figure 25.1 Structural Frame to Support the Test Equipment.

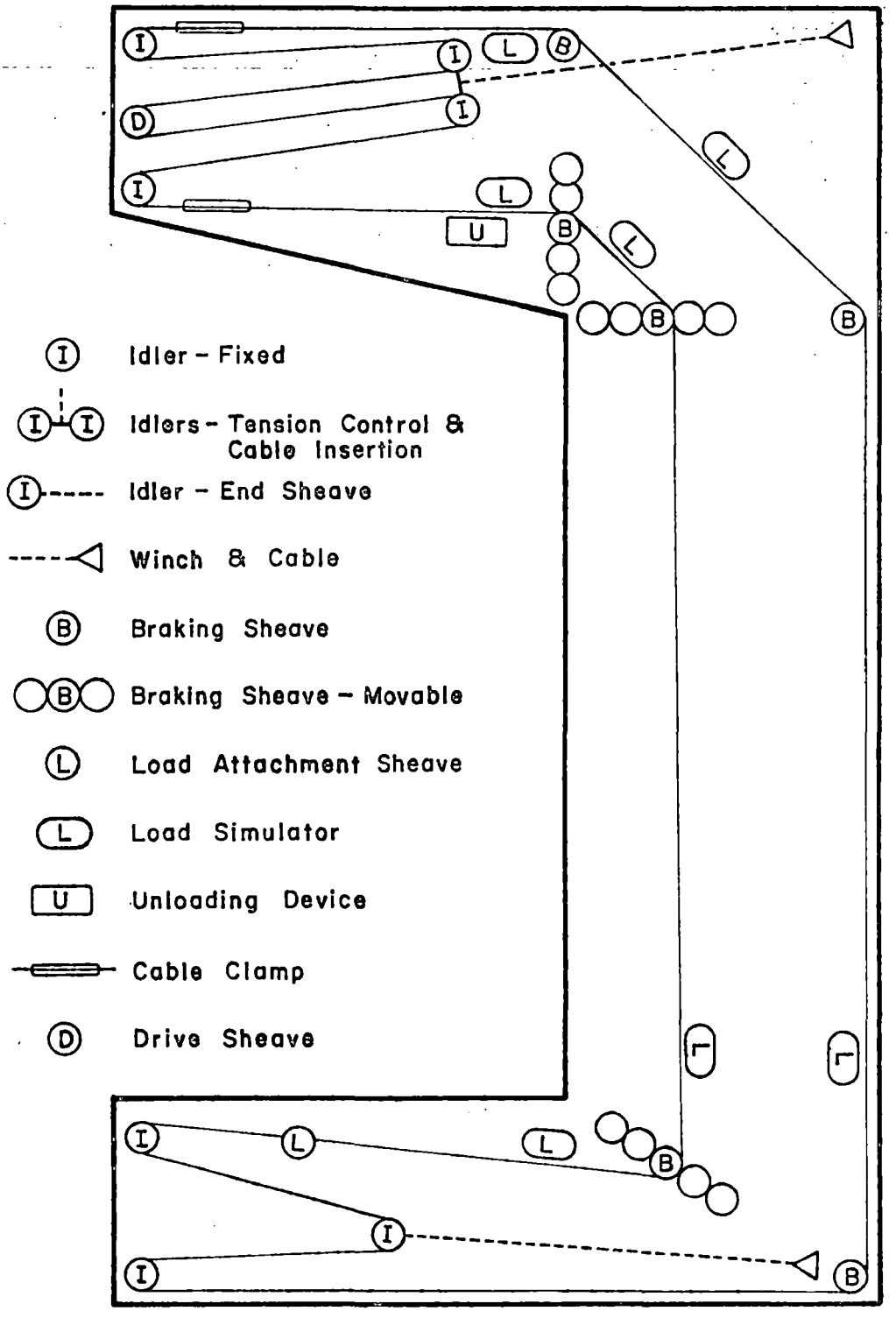


Figure 25.2 System Layout for the Test Facility.

The test facility will consist of the following items:

1. Three pair of braking sheaves.

The fixed sheave will be driven by the variable displacement motor and will simulate the control on the empty return line. The movable sheave will be driven by the fixed displacement motor and will simulate the control on the loaded incoming line. Moving the sheaves will permit tests on the angle of wrap and the effect of cable angle on the unloading device.

2. Four fixed idler sheaves to simulate idler support sheaves.

They also serve as cable guides for the cable tension control and cable insertion and removal device at the landing and for the end block in the woods.

3. A single idler sheave is connected to a hydraulic winch which will allow motion of approximately 18 feet or 36 feet of cable length. This simulates movement of the end block in the woods.

4. A double idler sheave is connected to a hydraulic winch with motion of approximately 18 feet or 72 feet of cable length change. This will be used to control cable tension and for the insertion or removal of cable segments. This will be installed near the support frame to carry the heavy cable loads. In the woods it would be near the ground to simplify cable changes.

5. The main drive sheave is shown between the two idler sheaves on the cable tension control and serves to increase the cable length change.

6. The load attachment sheave will pull the cable down to the top of the operator platform. It will be movable in or out from the wall to determine the effect of cable angle of approach on the loading device.

7. The unloading device will be tested with different cable angles of approach to determine its performance under variable conditions.

8. Seven load simulation devices will be used to simulate loads moving along the lines.

9. Two cable clamps will be used to hold the ends of the cable while it is disconnected for inserting or removing cable segments.

The profile of the truss frames are shown in Figures 25.3 and 25.4. The cable support areas will be 10 foot high. An additional 2 feet is used to hold the posts vertical and prevent buckling of the horizontal frame. The end frames 4 - 7 and 5 - 8 shown in Figure 25.3 drop to 8 feet, however, the cables will pass between these two frames at 10 feet. Strength analysis for the steel truss frame was based on the Eulers Column Failure Formula.

The calculations were based on rounded end columns. The minimum column size was for 2 1/2 inch pipes with a weight of 5.8 lbs./ft. Three inch pipe has a weight of 7.58 lb./ft. This will almost double the strength while increasing the weight and cost by 30%. Three inch pipe is still one size smaller than recommended by the Mechanical Engineering design tables. A welded assembly with fixed ends would double the strength and should be used when possible. The frame can be assembled in several sections outside the building, then moved into place. The total weight of the frame will be approximately 5800 pounds and will probably cost 40 to 50 cents per pound for materials.

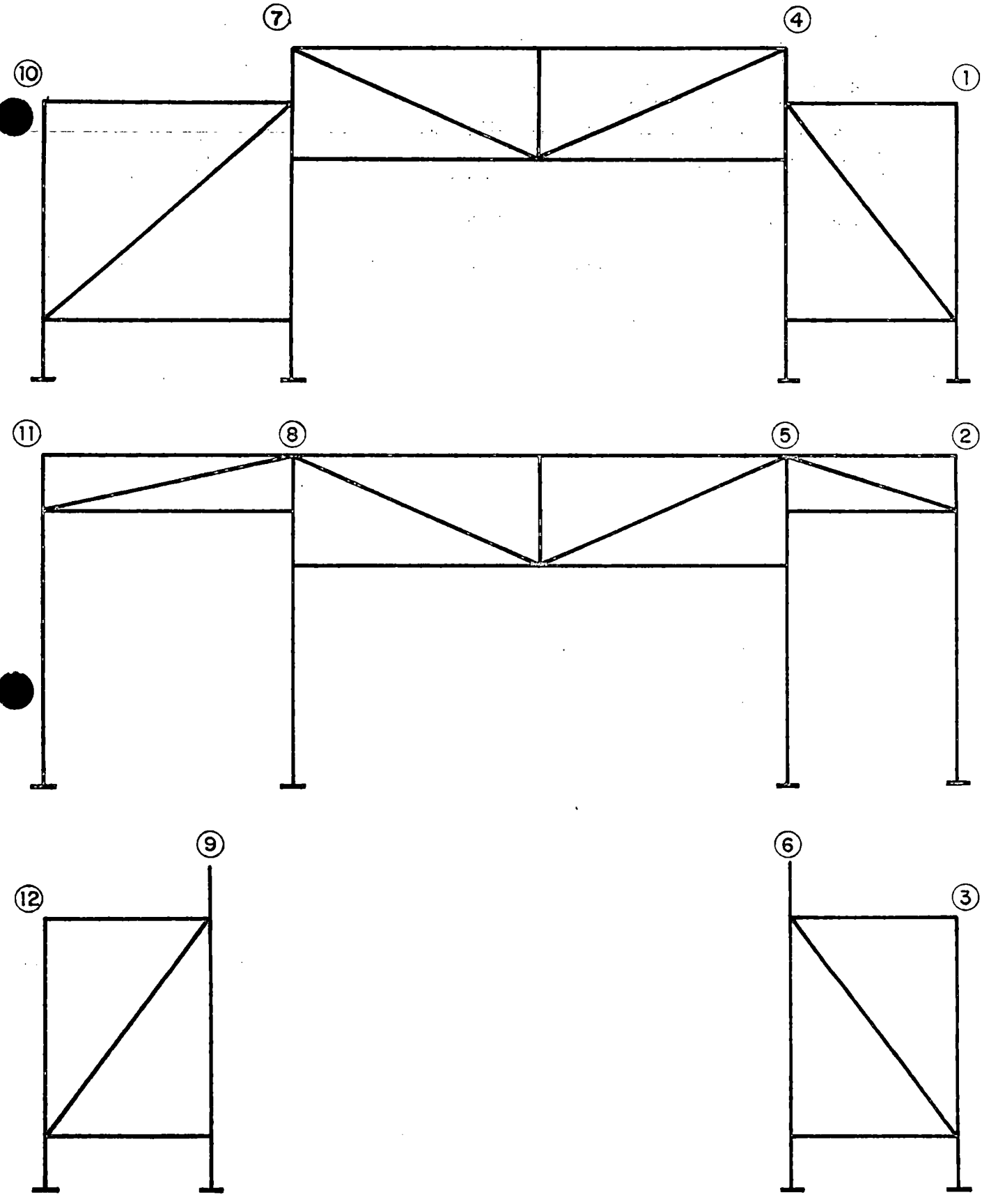


Figure 25.3 Profile of the North South Truss Frames.

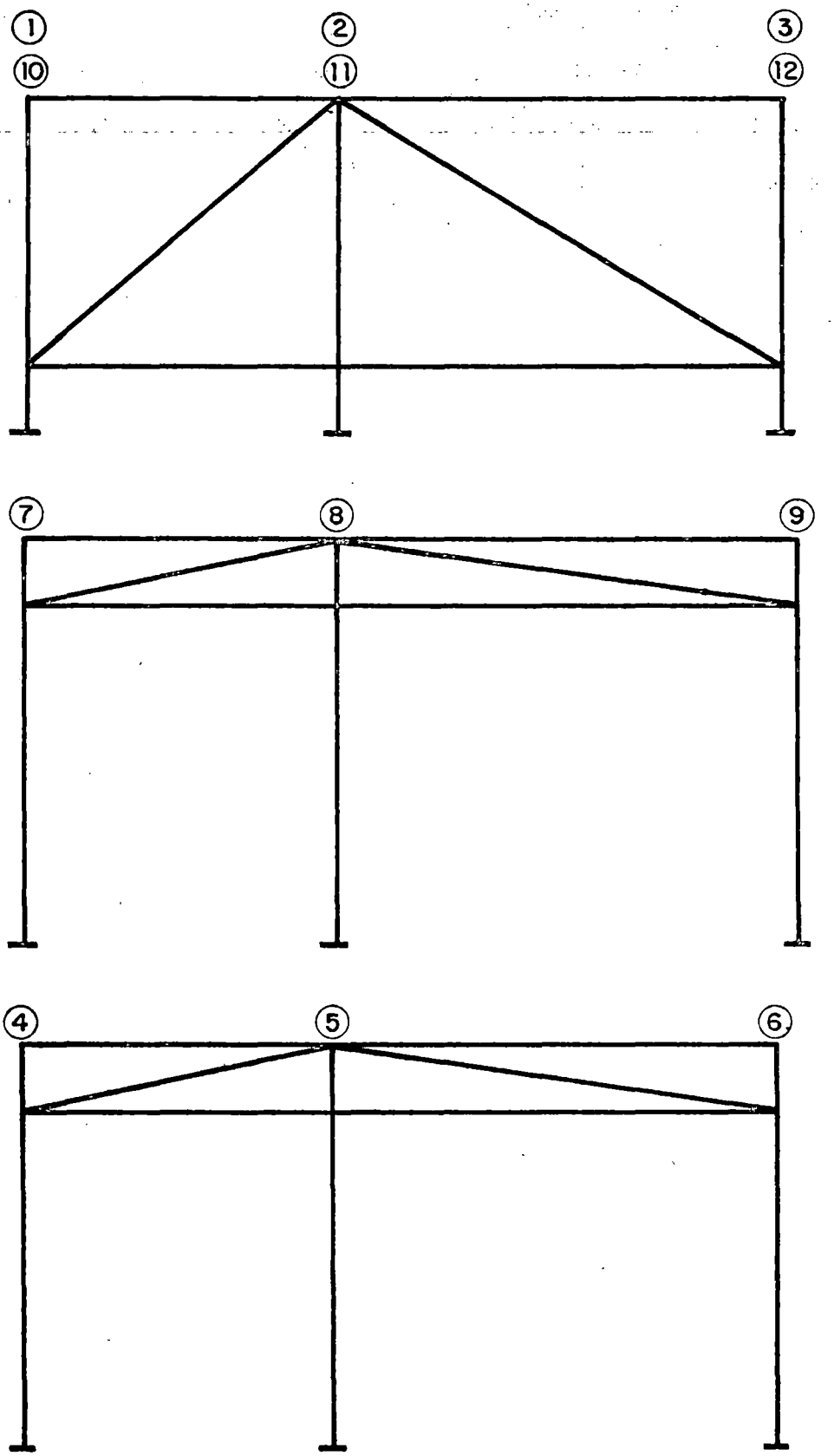


Figure 25.4 Profile of the East West Truss Frames.

Section 26. Develop a tentative test procedure for evaluating system performance.

The following steps will be involved in the test procedure for evaluating the system components and the performance of the total system.

a. Determine the loading conditions that will need to be simulated to duplicate field conditions.

b. Program and test a loading device to apply the design pulling or braking forces on the cable.

c. Apply known loads across a load control device and determine its response to the loads.

d. Apply known loads across a series of load control devices and measure the interaction between the devices.

e. Test actual loads on loading and unloading devices at low speeds.

f. Test techniques for the addition and removal of cable segments.

a. Determine the loading conditions that will need to be simulated to duplicate field conditions. *

The catenary cable equations were utilized in a computer program to determine the forces on a cable as a load passes from one support to the next. The equations were used to determine the cable tension differences that must be applied as braking or pulling forces on the cable to maintain control of the load. The braking forces were highest when the load was near the uphill support and lowest at the downhill support. The change in the control force as the load moved from one support tower to the next was very linear. These forces will

* The following is preliminary and tentative. A complete and accurate simulation model remains to be developed. Attachment 3 and 4 provide the basis for such a model.

therefore be simulated as straight line changes that can be solved much faster and easier than the catenary equations. The equations will predict the pulling or braking forces on the cable as a function of the position of the load between the support towers.

The simulation equations were a function of the slope, span length, uphill cable tension and the load. The position of the load within the span was a function of time and the cable velocity. The braking or pulling forces have a pattern similar to that shown in Figure 26.1 below.

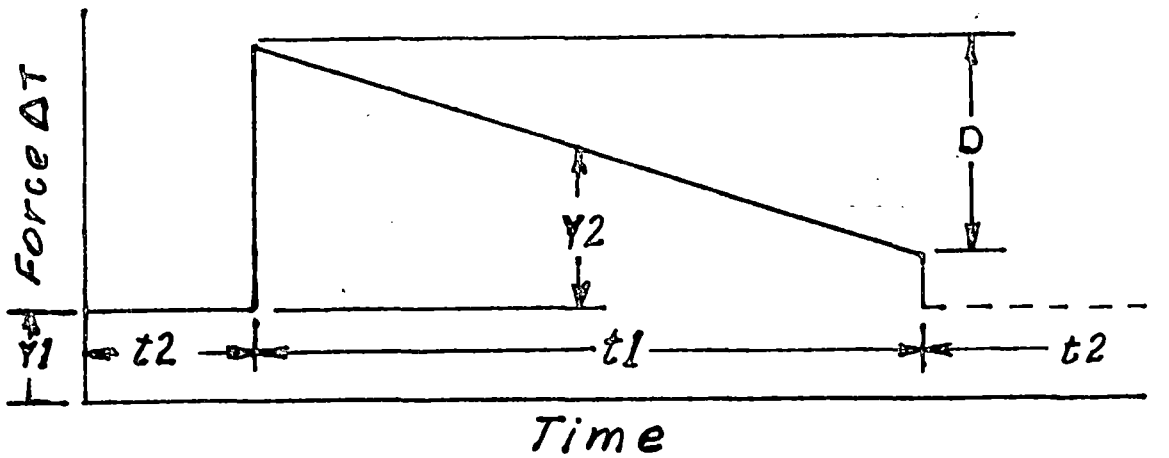


Figure 26.1 Typical Loading Pattern for a Load Simulation Device as the Load Moves Between Two Support Towers.

The symbols are defined as follows.

ΔT is the braking or pulling force that must be applied to the cable to duplicate or simulate field conditions.

Y1 is the load that is due to the cable alone.

Y2 is the mean value of the additional braking or pulling force that is due to the addition of a load to the cable. This mean value occurs at the mid point of the span because of the linear change between the supports.

D is the change in the braking or pulling forces as the load moves from the uphill tower to the downhill tower.

The total time cycle between loads is (t1 + t2). This time is a function of the length of the cable segments and the cable velocity, therefore:

$t1 + t2 = 500 \times 60 / Vel$

t1 and t2 are in seconds and the velocity is in feet per minute. The length of the cable segments is shown as 500 feet. Other lengths could be used.

$t1 = Span \times 60 / Vel$

t1 is the time in seconds required for the load to pass between two support towers. The distance is measured as the cord distance between the towers so that the time can be calculated directly as a function of the span and the velocity.

The force applied to the cable will be called ΔT and can be determined from by the equation:

$\Delta T = Y1 + Y2 + D/2 - Dt/t1$ for t = 0 to t1

$\Delta T = Y1$ for t = t1 to t2

The time cycle defined by these equations then repeats as a new load passes between the support towers.

The catenary equilibrium conditions were plotted on (x,y) coordinates then normalized to separate the effect of the cable size, the slope, the load and the uphill cable tension. The braking force due to the cable alone, shown as Y1 in Figure 26.1, is the product of the cable weight per foot and the elevation difference between the support towers. For 3/8th inch cable the equation is:

$Y1 = 0.259 \times Span \times Sin(Arctan(Slope/100))$

$$Sin(Arctan(Slope/100)) = \sqrt{\frac{Slope^2}{Slope^2 + 100^2}}$$

$$Therefore,\ Y1 = 0.259 \times Span \times \sqrt{\frac{Slope^2}{Slope^2 + 100^2}}$$

The mean value of the tension difference due to the load, shown as Y2 in Figure 26.1, is the component of the load support vector that is parallel to the cord.

$$Y2 = \text{Load} \times \sin(\arctan(\text{Slope}/100))$$

$$\text{Therefore, } Y2 = \text{Load} \times \sqrt{\frac{\text{Slope}^2}{\text{Slope}^2 + 100^2}}$$

The equation for the change in the braking or pulling force as the load moves across the span, shown as D in Figure 26.1, is a function of the slope, the span, the load and the tight side cable tension as follows:

$$D = 10 \left[61.16 \left(\frac{0.285 \times \text{Slope}^2 + 100^2}{\text{Slope}^2 + 100^2} \right) - \left(\frac{500 - \text{Span}}{40} \right) \right] \frac{\text{Load}}{\text{Tension}}$$

These equations can be used with given values for the span, slope, and cable tight-side tension to determine the braking or pulling forces needed to control different loads. The forces can then be applied to simulate field conditions as a function of time.

b. Program a loading device to apply the design braking or pulling forces to the cable.

This will be a test of the loading devices to determine how accurately they can duplicate the predicted field conditions. These test will be initiated using a hydraulic dynamometer as the energy absorbing device.

c. Apply known loads across a load absorbing control device and determine its response to the loads.

This will be a test of the braking devices. These devices must be able to measure cable tensions and speeds and control the cable tension by increasing or decreasing the cable speed. When cable tension is not limiting then the control device will maintain the desired speed.

d. Apply known loads across a series of load control devices and measure the interaction between the devices.

This test will simulate the total system so that interactions between devices can be determined. The simulated slopes, spans, and loads can be changed as desired to simulate different field conditions.

e. Test actual loads on loading and unloading devices at low cable speeds.

This test will be needed to develop and evaluate components such as chokers, latching devices and other attaching and detaching mechanisms. Considerable development and testing will be needed to perfect this untried technique for placing logs onto a moving cable.

f. Test the technique for adding and removing cable segments from the line.

This test will be used to develop a procedure for changing the length of the conveyor line and for controlling the tension on the line. A similar procedure would be used in the forest, however, the line segments would be much longer under field conditions.

All the test procedures will undergo changes as the tests proceed and new information is gained. This will be a dynamic relationship with results determining the need for modifications and additional testing.

Section 27. Design a data collection system for measuring system performance during the test sequence.

The purpose for data collection is to prove that the loading devices apply the desired loads and that the control devices respond properly to the loads. All measurements will be functions of time and will require simultaneous measurements at a number of locations. The quantity of data that could be collected could easily overload a computer data collection system. Information is needed to determine how accurately the system performs its intended function. Since this is a time function then a graphical representation will be most meaningful. Additional measurements that tell when the system is not performing as desired will also be needed. The graphical presentations will require data averaging or smoothing and a comparison of the actual and the desired results.

The data will need to be gathered by a number of different control processors under a master computer control due to the large number of simultaneous measurements that must be made. The data would need to be collected over the same time period at each location then forwarded to the master computer on request for comparison and analysis. The estimated collection time period for each set of data would be 2 to 5 minutes. This would allow 2 to 3 load cycles to pass over each control point. The load cycles would be repeated until equilibrium conditions were achieved after which data would be collected for the desired time period. This process would be repeated each time conditions were changed.

Section 28. Design a data analysis technique for visually displaying performance parameters and comparing them with design parameters.

Computer graphics will be used to visually compare the actual performance with the desired performance. The data will need to be stored so that a number a different measurements all taken at the same time period can be visually displayed in sequence. A statistical measure of the variation from the desired parameters will also be recorded for each parameter that is displayed.

Section 29. Determine a construction budget for the test facility.

The construction budget for the test facility was determined for a Grant Application to the USDA Competitive Research Grants Program for Forest and Rangeland Renewable Resources in Fiscal Year 1985 and for a brochure with the title, "Cable Conveyor Concept - A Research and Development Prospectus". A copy of the Grant Application and the Brochure are attached. The construction budget is estimated at $660,000 for a 40 month construction and test period. The budget includes a $500,000 grant request plus $160,000 in services and facilities from the USDA Forest Service Forestry Sciences Laboratory.

Section 30. Determine a construction schedule for the test facility.

The construction schedule for the test facility was determined for a Grant Application to the USDA Competitive Research Grants Program for Forest and Rangeland Renewable Resources in Fiscal Year 1985 and for a brochure with the title, "Cable Conveyor Concept - A Research and Development Prospectus". A copy of the Grant Application and the Brochure are attached. The construction time should require approximately 12 months out of the proposed 40 month construction and test period.

Section 31. Propose a test schedule for the forwarding system.

A test schedule for the forwarding system was determined for a Grant Application to the USDA Competitive Research Grants Program for Forest and Rangeland Renewable Resources in Fiscal Year 1985 and for a brochure with the title, "Cable Conveyor Concept - A Research and Development Prospectus". A copy of the Grant Application and the Brochure are attached. The test schedule covers a period of approximately 24 months during the 40 month development, test and report period.

ATTACHMENT I

GRANT APPLICATION TO THE USDA
COMPETITIVE GRANTS
FOREST AND RANGELAND RENEWABLE
RESOURCES IN FISCAL YEAR 1985

OMB Approved 0525-0001
Expires 12/31/86

UNITED STATES DEPARTMENT OF AGRICULTURE
SCIENCE AND EDUCATION

GRANT APPLICATION

FOR OFFICE USE ONLY	
PROPOSAL NO.	DATE RECEIVED
PROGRAM AREA CODE	PROPOSAL CODE

1. LEGAL NAME OF ORGANIZATION TO WHOM AWARD SHOULD BE MADE
Montana State University

2. ORGANIZATIONAL ADDRESS (Give complete mailing address & Zip code — include County)
Office of Grants and Contracts
Montana State University
Bozeman, MT 59717

3. NAME OF AUTHORIZED REPRESENTATIVE
Dr. John W. Jutila

4. TELEPHONE NO. (Include area code)
(406) 994-2891

5. ADDRESS OF AUTHORIZED REPRESENTATIVE (If different from item 2)
Vice President for Research
Montana State University
Bozeman, MT 59717

6. TITLE OF PROPOSAL (Maximum 80 characters)
An Endless Cable Conveyor System for Roadless Timber Harvest

7. PROGRAM TO WHICH YOU ARE APPLYING (Refer to Federal Register Announcement)
Competitive Research Grants Program for Forest and Rangeland Renewable Resources for Fiscal Year 1985

8. PROGRAM AREA (Refer to Federal Register Announcement)
1.0 Improved Utilization of Wood and Wood Fiber

9. IRS NO.
81-601045

10. CONGRESSIONAL DISTRICT NO.
1st

11. EXISTING S&E AWARD NO.
[X] NEW [ ] RENEWAL (Specify):

12. PERIOD OF PROPOSED PROJECT DATES
From: 6-1-86 Through: 9-30-89

13. NO. OF MONTHS
40

14. FUNDS REQUESTED FROM S&E (Form S&E-55)
$ 499,778

PRINCIPAL INVESTIGATOR(S)

15. P.I.#1 NAME (First, middle, last)
William E. Larsen

16. BUSINESS ADDRESS (Include Department & Zip code)
Department of Agricultural Engineering
Montana State University
Bozeman, MT 59717

17. P.I.#1 TELEPHONE NO. (Include area code)
(406) 994-2275

18. P.I.#2 NAME (First, middle, last)
W. Robert Taylor

19. P.I.#3 NAME (First, middle, last)

20. TYPE OF PERFORMING ORGANIZATION (Check one only)

- 01 [ ] USDA/S&E Laboratory
- 02 [ ] Other Federal Research Laboratory
- 03 [ ] State Agricultural Experiment Station (SAES)
- 04 [X] Land Grant University 1862
- 05 [ ] Land Grant University 1890 or Tuskegee Institute
- 06 [ ] Private University or College
- 07 [ ] Public University or College (Non-land Grant)
- 08 [ ] Private Profit making
- 09 [ ] Private Non-profit
- 10 [ ] State or Local Government
- 11 [ ] Veterinary School or College
- 12 [ ] Other (Specify):

21. Will the work in this proposal deal with recombinant DNA?
[X] NO [ ] YES

22. Will the work in this proposal deal with human subjects?
[X] NO [ ] YES (Complete Form S&E-662)

23. Will this proposal be sent to any other Federal Agency?
[X] NO [ ] YES (List acronyms)

CERTIFICATION AND ACCEPTANCE

We, the undersigned, certify that the information contained herein is true and complete to the best of our knowledge, and accept as to any grant awarded, the obligation to comply with the terms and conditions of Science and Education in effect at the time of the award.

SIGNATURE OF PRINCIPAL INVESTIGATOR(S)
William E Larsen William R Taylor
DATE 5/29/85

SIGNATURE OF AUTHORIZED ORGANIZATIONAL REPRESENTATIVE (Same as Item 3)
for Dr. John W. Jutila
TITLE Vice President for Research
DATE 5/30/85

Form S&E-661 (Aug 81)

(1) TITLE OF PROJECT: AN ENDLESS CABLE CONVEYOR SYSTEM FOR ROADLESS TIMBER HARVEST

(2) PROJECT SUMMARY:

Timber harvesting in the West has been and continues to be controversial. In particular, the roads needed to enable timber harvesting with existing logging systems, especially in steep terrain, are objectionable to many people. As a result a variety of skyline systems, balloons, and helicopters have been tried recently. The result has been more costly logging, but seemingly without reduction of controversy. This controversy appears to be a major factor limiting the future utilization of wood and wood fiber. Alternative systems to overcome this restriction are needed in order to utilize the forest resource.

This proposal concerns an ENDLESS CABLE CONVEYOR SYSTEM for harvesting timber, even in remote areas, without building a road network to support it. The overall objective is to design, develop, and test a roadless system for harvesting timber on relatively steep terrain that will be at least as cost effective as any systems in use today. Over two years of work has already been done investigating the technical feasibility of the concept. The tasks remaining include the following: final design of the indoor test facility, component procurement, fabrication of the test equipment and facility, conducting the indoor tests and collecting data, analysis of data, evaluation of the data and the overall design concept, system redesign or modification, and evaluation of the modified system. Once the concept has been proven in the lab, a small scale outdoor study will be run with actual logs. Future work will involve an in-woods site to evaluate a full scale system in a forest environment. The last phase of the project will be to make the research data available to equipment manufacturers in hope that this new technology could be transferred to the private sector as a cost effective approach to timber harvesting problems in the Intermountain West.

Since the timber industry and its related activities are presently depressed, this technology could help the economy by making it possible to harvest timber in inaccessible areas without building roads. Although the timber is relatively small (10-22 inches dbh) most of it is marketable. Presently much of it is maturing or susceptible to disease and will be wasted.

Preliminary investigation has indicated the cable conveyor concept has sufficient merit to proceed with its development, at least through a stage needed to demonstrate its technical feasibility. It appears to be at least as cost effective as any aerial systems presently being used. Preliminary economic analyses have indicated that this cable conveyor system will also compare favorably with present road-type systems. In addition, it will be more cost effective and practical than the heavy-lift, Heli-Stat type device presently being researched.

(3) PROJECT DESCRIPTION:

INTRODUCTION:

Timber harvesting in the Interior West has been and continues to be controversial. In particular, the roads needed to enable timber harvesting with existing logging systems, especially in steep terrain, are objectionable to many people. The forest industry recognizes that previous errors in timber harvesting have provided the examples for environmentalists and preservationists, with wide public support, to force changes in harvesting methods even though they may not be the most economical (3, 4). Forest work roads are the most visible portion of a logging site and the public has associated these roads with environmental damage. This damage has been associated with detrimental impacts on soil and water resources. The process of opening the wild land to public access, more specifically motor vehicle access, has often been considered to be as harmful as the soil and water problems (1, 4, 12, 16, 17, 21, 22). In addition to public concern, the Oregon Legislature directed the Oregon State University, School of Forestry, to seek solutions to the problems associated with roads (4).

In addition to environmental issues the forest industry must face the problem of an increasing portion of the available timber being of smaller size located on difficult terrain. This terrain has steep, sensitive slopes that require expensive roads and harvesting techniques. Alternative harvest methods are needed to alleviate environmental concerns if this resource is to be harvested (6, 8).

In response to the controversy surrounding road construction and logging in mountainous terrain, industry and government agencies have cooperated in recent decades to reduce the density of roads by using skyline systems, balloons, and helicopters. In addition, the environmental impacts of roads have been mitigated through more expensive construction methods. The result has been more costly logging, but seemingly without reduction of controversy. Indeed, those who object to the environmental consequences of logging in mountain terrain are now complaining about the financial consequences as well. The high cost of roads in difficult terrain has required a tax subsidy for road construction, or an expenditure that may not be recovered from the logging operation (16, 22).

The logger has long dreamed of an easier way to harvest timber. The ultimate system is visualized as a "sky hook" that would pick up logs from distant hillsides and deposit them gently on the valley bottom (1, 8). These dreams must be tempered by economic reality and partial approaches to this ideal system have included different cable systems, balloons, helicopters and heavy-lift airships. Many studies have been made to compare the economics of different systems and there effect on the amount of harvest work roads required (4, 20). These have included models to optimize road design and to minimize environmental impacts (2, 6, 13).

The transportation of logs from woods to mill involve many different terrain and surface conditions. The most difficult and expensive portion on a ton mile basis is the stump to landing portion. This portion therefore deserves and has received a great deal of research and development effort (21).

Cable systems were developed as a method to harvest steep slopes with a minimum of expensive roads. These systems have been described by names such as highlead and skyline. Their intention has been to partially or completely suspend the log so that they can be hauled into the landing with a minimum of environmental damage. Even so, there is damage to residual stands and other resources unless the logs are lifted clear of the ground. Downhill and sidehill yarding is particularly damaging when the logs touch the ground. These systems are generally limited to a yarding distance of 500 to 1500 feet. The distance can be increased by the use of heavier cables and intermediate supports, but the cost increases rapidly and production decreases with increased length (1, 3, 4, 7, 13, 20).

Balloons in various sizes and shapes have been used to extend the range of cable yarders. They have given good control of environmental damage but the weight of the pulling cables and the time required to travel long distances has limited thier use. Yarding distances are usually limited to 3000 to 5000 feet. High speed, hydrostatic winches with cable speeds up to 3000 FPM are required for good performance. Balloon use is restricted under conditions of snow, ice and wind. Weather may be one of the most restricting elements to aerial systems (1, 4, 20).

Helicopters have been characterized as low lift aircraft. Their ability to operate over difficult terrain without ground control make them an ideal logging vehicle. The high operation and maintenance cost of helicopters and their reduced productivity as yarding distances increase has limited yarding distance to approximately 5000 feet. Weather hazards, particularly wind gusts in mountainous terrain, have removed most of the low cost surplus helicopters from the woods (4, 5).

Heavy-Lift Airships that combine the lifting ability of balloons with the maneuverability of helicopters have been sought as a way to reduce logging costs in mountainous terrain. A considerable amount of money has been spent to develop these airships with the emphasis being on the Cyclo-Crane and Heli-Stat. It is hoped that these vehicles can profitably extend aerial yarding distances to 5 miles. The logistics of these machines is very critical as they must operate at a very high capacity to repay the high investment and operating costs. These machines are very dependent on favorable weather and must be tied down when winds exceed 25 MPH. The first Cyclo-Crane was destroyed in a coastal storm before any field test could be performed. A second model has been constructed and successfully field tested. The service area for the Heli-Stat requires a 7 acre circle within 5 miles of the logging site. The Heli-Stat has not been tested to date (11, 14, 21).

Another alternative developed for wet swampy conditions was a movable temporary roadway. This monorail system was tested for Muskeg conditions. It was a train type system that could be installed in and removed from the wet Muskeg areas. The train was limited to 2 percent slopes and was therefore not adapted to mountainous terrain. Rubber tires on the rail could increase the tractive effort but slopes would still be limited to 8 percent or less (21).

Conveyors are recognized as an alternative to roads and aerial systems. They are capable of traversing difficult terrain and long distances. The capacity of a conveyor is not affected by increased yarding distances because it is a continuous flow process. A major limit to conveyor systems has been their high initial cost and the difficulty of installing and moving them (21).

Low cost conveyors that use suspended cables to carry the logs out of the woods are needed and are being developed and tested. The Japanese ZigZag system is the most successful system at this time. They manually lift logs to the cable and then attach them with one or two rope chokers. At the landing the logs are released by a knife that automatically cuts the rope chokers. The ZigZag pattern pulls the open sided sheaves to one side so that the chokers can pass by the sheaves. Cable tension is controlled by a movable sheave on a winch cable that pulls slack out of the line between two support points. The Japanese ZigZag system is limited to small timber that can be manually loaded and yarded over relatively short distances. System control is limited to one main drive position and the cable tensions cannot be adjusted at intermediate points (15, 23).

An alternative cable conveying system has been proposed by the authors of this proposal. This alternative introduces hydrostatic drives with microprocessor monitoring and control at intermediate points along the line. This system will be able to mechanically load and remove logs weighing up to 2000 pounds from the moving cable. Cable speeds up to 500 FPM are relatively low, as compared to highlead or balloon systems, but one load of 2000 pounds every 500 feet on a cable traveling 500 feet per minute will deliver 60 ton per hour to the landing. Cable segments, 500 feet in length, can be easily added or removed from the system to facilitate rapid change in the systems dimensions. The mechanical power to install and change the system will be obtained from Mountain-Climbing Backhoe type machines. These machines were designed to operate on steep terrain for construction jobs typical of the tasks required for erecting and moving the conveyor. The Mountain-Climbing Backhoes are not adapted to tasks that require frequent long distance moves as the are very slow under these conditions (17, 19). Recent investigations by Michael Gonsior, the Forest Service Research Engineer collaborating on this project, have shown that the Mountain-Climbing Backhoe can be used successfully in a yarding and decking operation typical of that required to bring logs to a cable conveyor line.

Faculty and graduate students at Montana State University have been associated with various forms of logging research for a number of years (2, 5, 6, 8, 9, 12, 17). The earliest work focused on evaluating existing logging systems and developing plans and models for more efficient timber harvest. The current proposal has been under study for two and one-half years and has been funded by the USDA Forest Service, Intermountain Forest and Range Experiment Station. During the first year of the project, new, innovative alternatives were proposed and evaluated to determine the best approach to solving existing problems. Emphasis was placed on timber harvest problems of most concern in the Intermountain West. These steep mountainous slopes with relatively small timber in a dead or dying state are also typical of many areas throughout the world. The concept of AN ENDLESS CABLE CONVEYOR SYSTEM FOR ROADLESS TIMBER HARVEST appeared to have the greatest potential for these sites (10). Many of the limitations and problems associated with conveyor systems (18, 21) would have to be overcome to make the system economical. Concepts were developed to construct a system using relatively light weight cable as a log suspension system. The cable would be supported from existing trees or temporary towers. The conveyor could be easily and rapidly moved by adding or removing preassembled cable segments (500 foot lengths). This proposed system was compared with current systems and proposed new systems such as heavy lift airships. These airships are still not available for test purposes although a considerable effort has been made to develop such capability (11, 14, 20).

A favorable response from the Forest Service sponsors led to a second year of research where the details of a cable conveyor system, the control functions and the desired operating parameters were determined. The zigzag layout for the proposed concept is shown in Figure 1. Preliminary concepts were developed and a proof of concept test facility proposed. The need for a test facility to determine the design and operating properties for the individual components and the total system was seen as the essential next step in the development process. These design and operating parameters can be most economically determined under laboratory conditions where the forest operating loads and cable tensions can be simulated under controlled and monitored conditions (17).

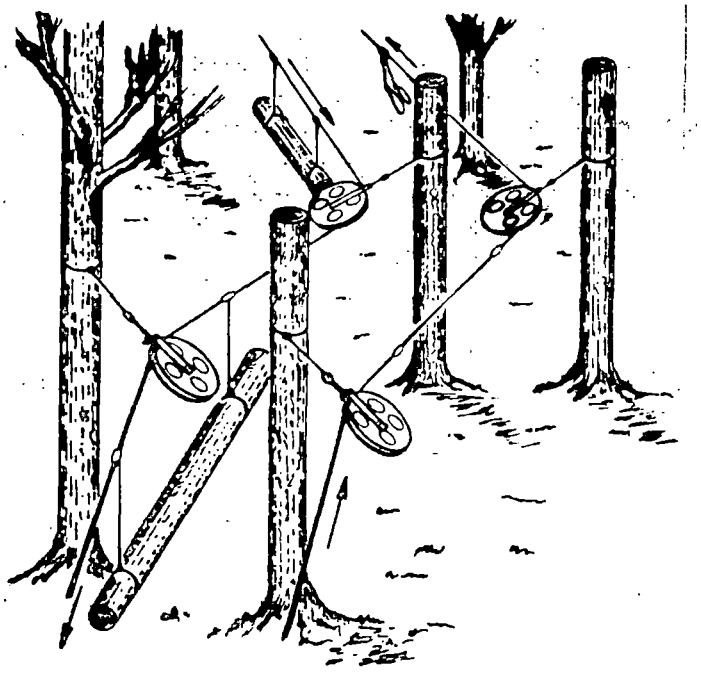


Figure 1. Zigzag Layout for the Endless Cable Conveyor System

The long term goals of the proposed research are to provide an alternative harvesting system for fragile and steep terrain that is competitive with existing systems and proposed heavy lift airships. The more specific goals are as follows:

A. To provide an alternative logging system for timber harvest in steep fragile terrain.

B. To provide an economic alternative to extensive work roads for timber harvest.

C. To provide a logging system that is compatible with environmental concerns relative to maintenance of soil and water resources.

D. To provide a logging system that utilizes a continuous flow conveyor system that will maintain capacity with increased yarding distances. (Distances up to five miles or more are envisioned)

E. To develop a logging conveyor system that can be readily extended and moved to accommodate different logging sites and conditions.

F. To develop a logging system that utilizes computer control technology to minimize the load variation over the length of the system and thus permits increasing the length of the system without developing excessive tensions that will require larger and heavier cables.

G. To develop a logging system that can be readily extended or moved to permit timber harvest over extended yarding distances without loss of system capacity.

RATIONALE AND SIGNIFICANCE:

The proposed research project, AN ENDLESS CABLE CONVEYOR SYSTEM FOR ROADLESS TIMBER HARVEST, appears to have high potential for solving many of the timber harvest problems in the Intermountain West. It also has application to many other areas with difficult harvest conditions and environmental restraints. The next step in the research, and the portion covered in this proposal, is to develop a Proof of Concept Test Facility and perform tests to prove that the proposed system can be made and operated using existing components and control technology.

The research proposal specifically applies to the research areas listed in the Friday, April 26, 1985 Federal Register / Vol. 50, No. 81. This particular project will help in the utilization of the forest resource by permitting harvest of these resources in spite of current and foreseeable future restraints. The specific area of research to be supported in fiscal year 1985, as described in the Federal Register is:

DEPARTMENT OF AGRICULTURE
Competitive Research Grants Program
for Forest and Rangeland Renewable
Resourses for Fiscal Year 1985;

1.0 Improved Utilization of Wood and Wood Fiber

This project will address a critical barrier to improved wood utilization, namely the increased difficulty of obtaining adequate supplies of wood from the forest due to logging restrictions caused by public concern, more difficult logging sites and environmental constraints.

The specific problems that must be solved require the development of a proof of concept test facility so that the system design and operating parameters needed in the forest can be developed and tested under controlled laboratory conditions. The research will require a test facility and the performance of a series of tests that simulate known forest conditions. The proof of concept test facility has been designed and the required tests determined. The test facility will be constructed in the USDA Forest Service Forestry Sciences Laboratory on the Montana State University Campus. A 60 foot by 40 foot building is available for this use and the west portion has been designated as the location for the test facility. An adjacent outside area that was formerly used for wood chip pipeline studies will be available for outdoors studies when actual logs are run through the system at full speed to test the mechanical loading and unloading devices. The test facility will use a one-half size cable (3/8 inch) to permit use of standard cable components which will result in design loads of one-fourth those expected in the woods. Cable speeds will be the same as those expected in the woods so the power level will also be one-fourth that expected in the woods.

The plan for the test facility is shown in Figure 2. A steel frame inside the Forestry Sciences Laboratory shop will simulate the tree and tower supports in the woods. Sheaves with an open side will be suspended below the supports, the zigzag pattern will pull the sheaves to one side to allow the loads to pass around the sheaves. Cable connectors will permit rapid addition or removal of cable segments to simulate moving the cable in the woods. The connecting links also serve as load supports to avoid point loads on the cable. There are six

braking sheaves that are used to duplicate the speed and load control functions that will occur in the woods. These sheaves with hydrostatic drives, tension and angle sensors and computer controls will simulate the control devices that will be needed on most support points in the woods. Seven load simulators will apply predetermined loads to the cable to simulate the changes in cable tension that would be caused by a load moving between two supports. These pulling or braking forces are a function of load, span length, slope and cable tension. A computer will control these loads as a function of cable speed and will be able to duplicate predetermined field conditions. Different load cycles will be programmed as needed.

A single variable speed drive unit will be the master speed control for the system with the rest of the system responding to this drive to maintain the same speed without overstressing the cable at any point along the line. The cable take-up near the drive will control cable tension and will be used to simulate the process of adding or removing cable segments. The cable take-up at the other end of the loop will simulate the movement of the cable in the woods.

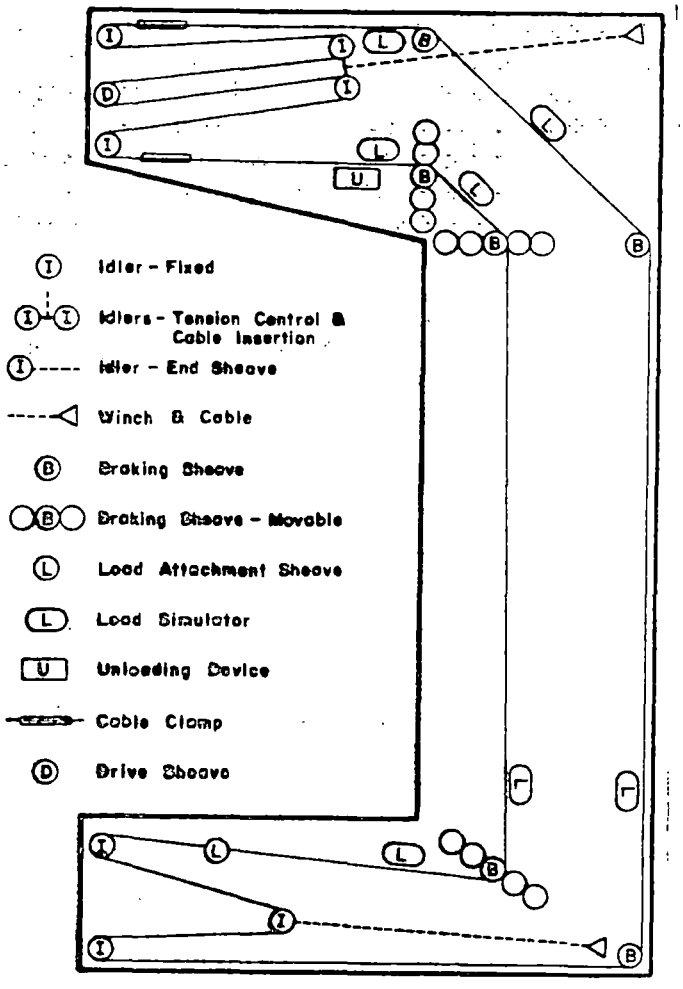


Figure 2. Plan View of the Proof of Concept Test facility

The specific test that must be run to prove the cable conveyor concept and to develop operating guidelines for a full scale system are:

1. Prove that the load simulation device can apply the design force on the cable to accurately simulate the loads that will occur as a log moves between two support towers. The sheave torque will be monitored and compared to the desired load as determined by the control computer.

2. Prove the effectiveness of the controlled sheaves in maintaining the cable tensions within design limits. The actual cable speeds, tensions and sheave torque will be monitored and controlled.

3. Design and test the procedure for adding and removing cable segments.

4. Develop and test automatic load connection and removal devices and test their performance. Tests will be limited to simulated loads at full speed and actual logs will be used indoors only at low speeds.

5. Perform outdoor tests to reevaluate the loading and unloading systems with actual logs at full system speed.

EXPERIMENTAL PLAN:

The experimental plan consists of four phases which are depicted in Figure 3. The preliminary design and analysis has already been completed under a Cooperative Agreement between the USDA Forest Service Intermountain Experiment Station and Montana State University. This proposal concerns Phase II, Proof Of Concept, in which the operational, safety, and design limits would be established via performance tests in a 1/2 scale indoor test facility. The objectives of this phase are to acquire production, system performance, and cost

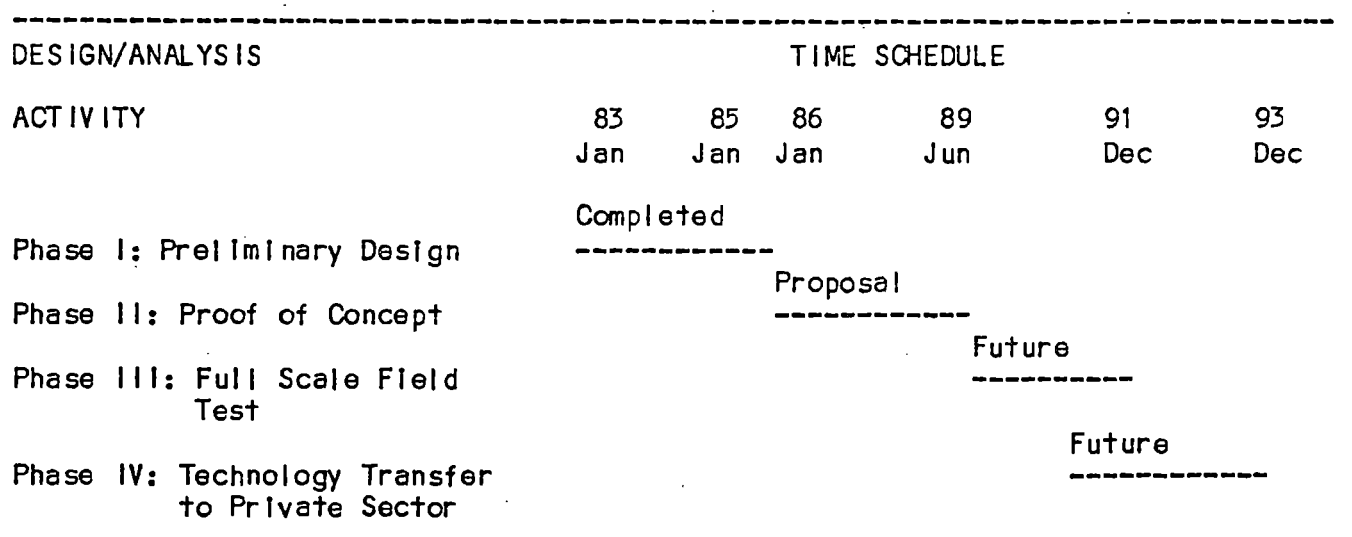


Figure 3. An Overview of the Experimental Plan

data to document the viability of a full-scale system in the woods. Phase II will provide an opportunity to validate each subsystem, verify system performance, as well as data collection and control systems. The technical feasibility of the overall concept and equipment designs will be monitored and analyzed. More detail regarding the sequence of activities is included in the Tentative Schedule.

It is expected that Phase II will likely result in some design modifications being required. Indeed, one of the purposes for qualification testing is to identify design or operational problems prior to manufacturing the equipment. Formal design reviews will assist in identifying, documenting, and eliminating such problems.

The system performance will be monitored and data will be collected via sensors and microprocessors. Experiments will be statistically designed in order to facilitate explanations of variations in system performance. Prior to running any test, the experimental model will be defined so that interpretation of the results may be statistically documented. The objective here is to transform data into information and knowledge.

Research necessarily involves risks. Several pitfalls and limitations are possible with this research. Probably chief among them is the assumption that

all will go well, as planned. It will not. It never does in any creative activity. Attaining the desired result in accordance with a schedule is very difficult and will require timely identification and resolution of bottlenecks, and regular, results-oriented design and performance reviews. Normal delays will occur due to equipment failure, shipping delays, etc. The tentative schedule which follows anticipates many of these problems.

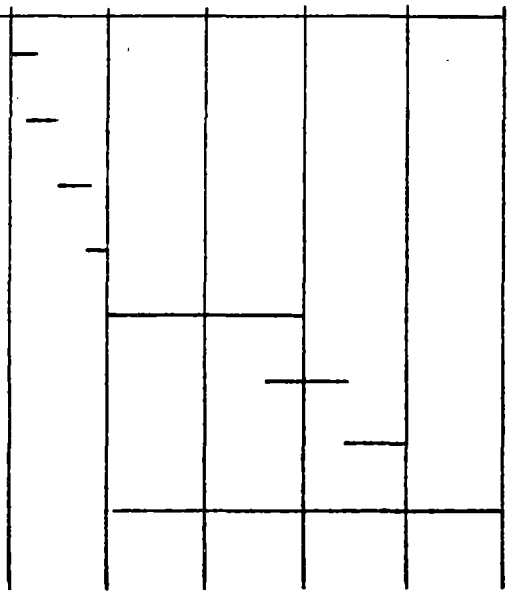
Once the concept of an endless cable conveyor system (ECCS) has been through a test facility "shakedown" and qualification, it is planned to fabricate and assemble a full-scale version for field trials in the woods (Phase III). In Phase III, the ECCS will be tested under a variety of conditions via many operational policies and methods. Operational limits will be identified and methods for optimizing productivity at least costs will be examined. Extensive cost and productivity data will be acquired for analysis.

Upon completion of Phase III, it is anticipated that a program will be required to facilitate technology transfer to the private sector. Phase IV is designed to encourage manufacturers to produce and market this new technology. Tentative schedules for all phases of this research follow.

PHASE I
Completed: June 30, 1985

Design/Analysis Activity

Time Schedule (Months)
3 6 9 12 15 18 21 24 27 30



Problem Definition

Alternative Solutions

Evaluate Alternatives

Select Solution Concept

Refine ECCS Concept

Technical Feasibility

Economic Feasibility

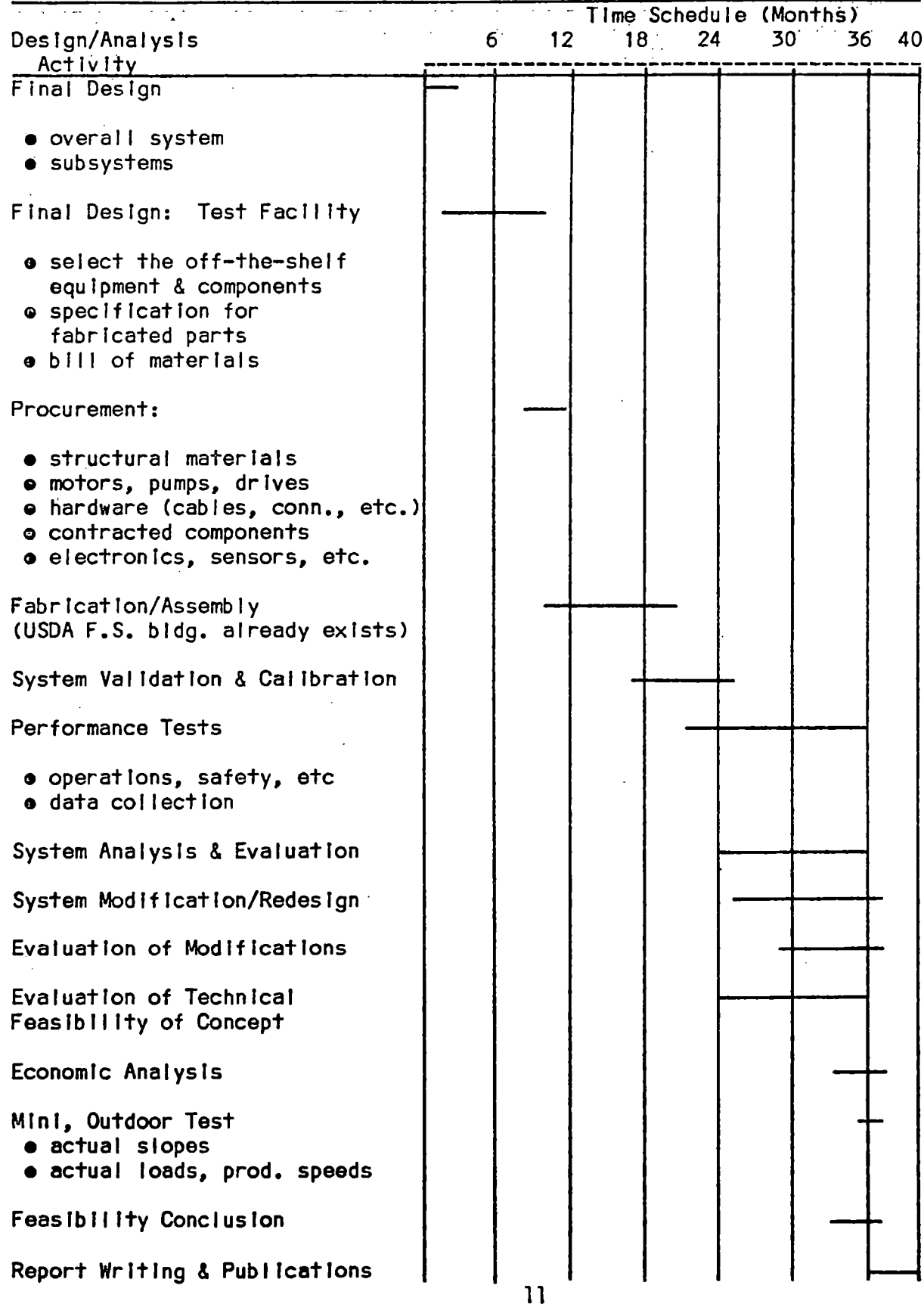
Preliminary Design of System Components:

- cable segments, connectors, speeds, loads, tensions
- sheaves, pass-through cable connectors
- hydraulic motors, gear reduction, bearing design
- sheave support frame
- sheave support tension sensor
- angle sensors re cable
- cable speed sensors-operating parameters
- hydraulic motor speed control
- hydraulic motor displacement control

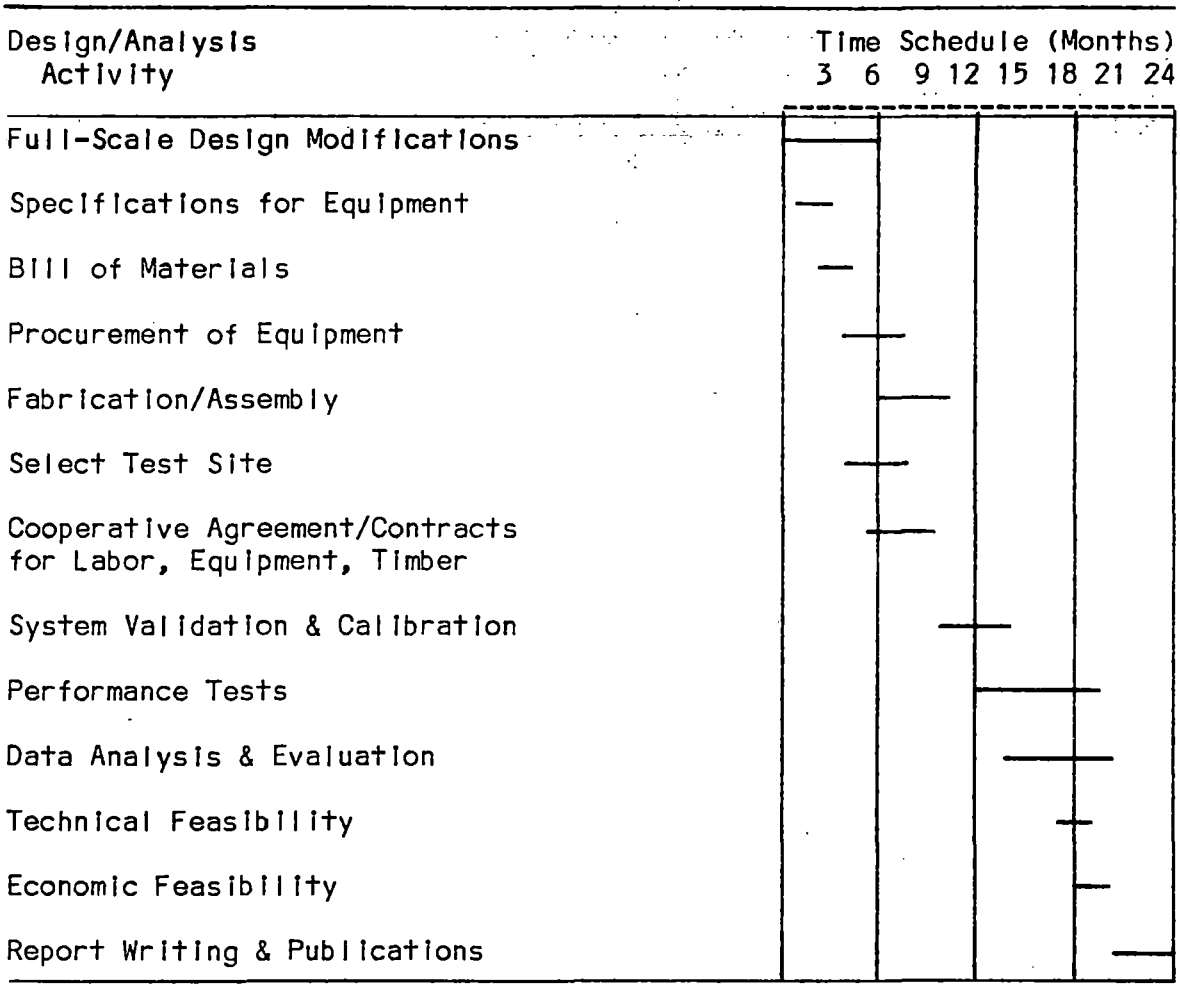
PHASE I (continued)
Completed: June 30, 1985

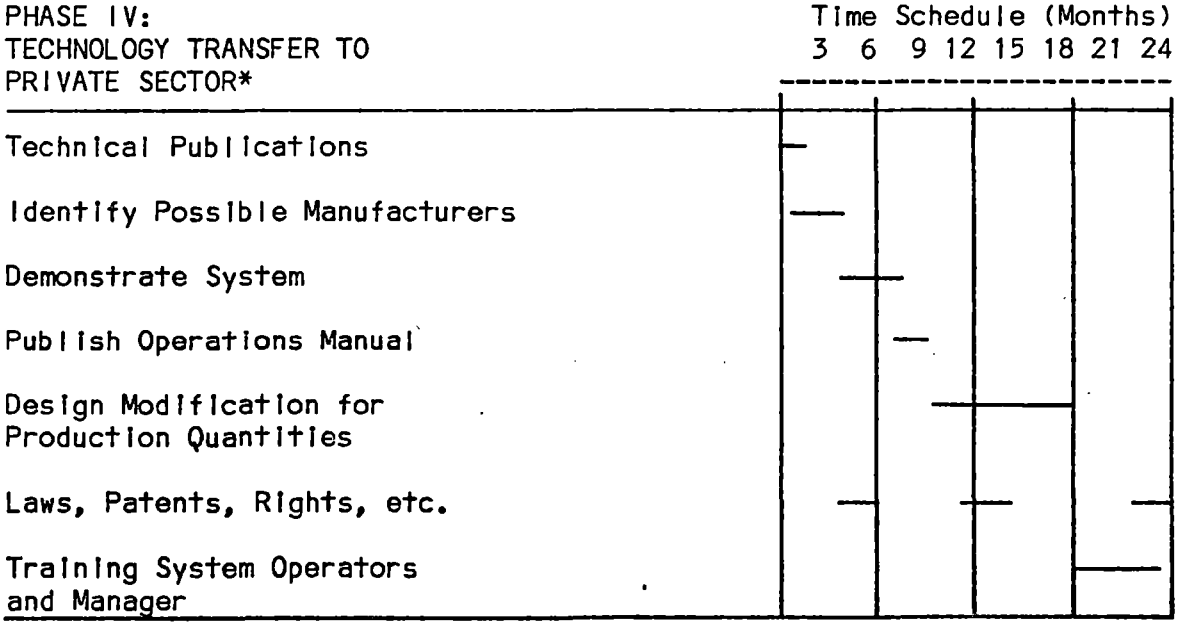
Design/Analysis Activity	Time Schedule (Months) 3 6 9 12 15 18 21 24 27 30
• pressure relief valves	
• charge pump capacity for anti-cavitation control	
• filter & reservoir	
• heat exchanger	
• alternator, regulator, battery, etc.	
• pump & engine for hydraulic power supply	
• pump displacement control (pressure and speed)	
• loading device and simulation methods	
• end sheave design: mounting & cable take-up	
• idler sheave design	
• main drive unit (speed & torque control)	
• loading unit	
• unloading unit	
• cable tension control device	
• adding/removing cable segments	
• cable support frame	
• test plan design	
• data collection system	
• system analysis	
• test facility	
• test schedule	

PHASE II: PROOF OF CONCEPT
TEST FACILITY



PHASE III:
FULL SCALE FIELD TEST





*Manufacturers, USDA Forest Service Equipment Development Center, etc.

FACILITIES AND EQUIPMENT:

A number of laboratory and shop facilities at Montana State University have been reserved for this research project. The actual test facility will be constructed and the tests run in the USDA Forest Service Forestry Sciences Laboratory shop. A 40 foot by 40 foot space in the west end of the building will be used in this manner. A large door leads directly to the narrow but 1200 foot long outdoor test site which was previously used for testing a wood chip pipeline. This will permit use of the same power source and instrumentation in both test locations. The Agricultural Engineering Research shop will be available for the fabrication and assembly of components and the Mechanical Engineering machine shop will be used for the fabrication of components requiring numerically controlled machine tools.

The computer facilities available for control, data collection and data analysis will consist primarily of an AT&T 6300 microcomputer with 640K of RAM, a 10 Mb hard disk and a floppy disk drive. Programs for data analysis that are too large for this system will be run on the Honeywell CP6 or one of the VAXs. The AT&T 6300 was purchased specifically for this project.

The non-expendable equipment needed for this project will include the mechanical drives, the hydraulic components, the instrumentation transducers and the computer interface devices. The cables, chokers and support frame will be considered as expendable items. A detailed list of this equipment and the estimated cost is shown on the budget sheet.

COLLABORATORS

The USDA Forest Service Research Engineer and the Electronics Technician who have been involved in the project for the last two years will continue to work on approximately a half time basis on this project. The primary collaborator will be Michael J. Gonsior, USDA Forest Service Research Engineer who initiated the cooperative study in which the concept and initial design was developed. Mr. Ron Babbitt, a USDA Forest Service Electronics Technician working for Mr. Gonsior, will also assist in the development of the control and operating system. Other USDA Forest Service personnel at the Forestry Sciences Laboratory in Bozeman and elsewhere in the Intermountain Research Station will be available on a consulting basis. The USDA Forest Service support on this project will be provided at no charge to the research grant.

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11. Gulmier, D. Y. and G. V. Wellburn, Logging With Heavy-Lift Airships, FERRIC, Forest Engineering Research Institute of Canada, Technical Report No. TR-58, May 1984.

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(5) VITA OF PRINCIPAL PROJECT PERSONNEL

The following resumes are attached:

William E. Larsen, Co-Principal Investigator

William R. Taylor, Co-Principal Investigator

Michael J. Gonsior, Principal Collaborator

RESUME

William E. Larsen Born: August 21, 1928, Hyannis, Nebr.

Education:

1946-1948 Doane College, Crete, Nebraska
Pre-Engineering

1948-1950 University of Nebraska, Lincoln, Nebraska
Degree: B.S. in Agricultural Engineering

1954-1957 University of Delaware, Newark, Delaware
Degree: Master of Mechanical Engineering
Thesis: The Design and Application of a Toothed Reel for a Lima Bean Harvesting Machine

1965-1968 University of Illinois, Urbana, Illinois
Degree: Ph.D. in Agricultural Engineering
Title of Dissertation: A Control System for Off-Highway Vehicles

Professional Experience - Montana State University:

1968-1972 Associate Professor of Agricultural Engineering. Teaching and Research joint appointment in the College of Engineering and the College of Agriculture, Agricultural Experiment Station.

1972-1973 Professor of Agricultural Engineering. Acting Department Head for two quarters. Promoted to Head of the Agricultural Engineering Department in 1973.

1973-present Professor and Head of Agricultural Engineering.

Professional Experience - Other Related Experience:

1950-1951 University of Nebraska, Tractor Testing Laboratory, Lincoln, Nebraska. Assistant Tractor Test Engineer.

1951-1954 U.S. Public Health Service Commissioned Officer in the active reserves. Assistant Sanitary Engineer.

1954-1959 University of Delaware, Newark, Delaware. Instructor and later Assistant Professor. Research, Teaching and Extension.

1959-1967 University of Arizona, Tucson, Arizona. Extension Agricultural Engineer, Power and Machinery. On sabbatical leave 9-65 to 9-66 at the University of Illinois.

1967-1968 University of Illinois, Urban, Illinois. Research Associate. Completed requirements for a Ph.D. In Agricultural Engineering.

Consulting:

Consulting is quite limited with one to two assignments of one to four days each per year. Consulting involves machinery accidents, machinery performance and machinery management.

Professional Registration:

Registered Professional Engineer in Arizona and Montana. Registered in both Agricultural Engineering and Mechanical Engineering.

PUBLICATIONS LIST

Ohrmann, J.O, W.E. Larsen, and L.R. Erickson, "Four Wheel Drive Tractors: A Field Evaluation," Agricultural Engineering Journal, V. 61, No. 12, pp 17-18, 1980.

Larsen, W.E., "Efficiency of Utilization of Four-Wheel Drive Tractors," Refereed proceedings of the ASAE Energy Symposium, Kansas City, Missouri, 1980.

Larsen, W.E., "Draft and Energy Requirements for Tillage Operations," Proceedings of the North Dakota Tillage Symposium, Bismarck, ND, 1980.

Erickson, L.R., W.E. Larsen and J.O. Ohrmann, "Efficiency of Four Wheel Drive Tractors," Montana Tillage Conference, Great Falls, MT, March 9-11, 1981.

Erickson, L.R., B.A. Ekholt, W. Nixon and W.E. Larsen, "AGNET Computer Program MACHINESIZE," Agnet, 319 Linfield, Montana State University, Bozeman, Montana, 1981.

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Larsen, W.E., "Equipment Requirements for Reduced and No-till Grain Production," Proceedings; Great Plain Conservation Tillage Symposium, Publication 110, 1984.

Larsen, W.E., M.J. Gonsior, and W. R. Taylor, "A Cable Forwarding System - Test Facility Design," ASAE Paper 84-1616, 1984.

Larsen, W.E. and D.G. Miller, "Alternative Diesel Fuel Production of Oilseed Crops for Northern Climates, Seminar Summary," Montana State University, March, 1984.

Larsen, W.E., "Alternative Diesel Fuel Vegetable Oils In Montana," Summary of the Data from the Goering, Eslick studies in the 60's, Montana State University, 1984.

Larsen, W.E. and A.R. Rider, "Mechanization of Forage Harvesting and Handling," Chapter 48, Forages, 4th Edition, Iowa State Press, 1985.

Ismail, S.M., G.L. Westesen and W.E. Larsen, "Surge Flow Border Irrigation Using an Automatic Drop Gate," Transactions of the ASAE, 1985.

Professional Membership:

Member of ASAE since 1950, served as chairman of the Arizona section section in 1964. Served as secretary of the Montana section for two years, served as chairman of the PNW region in 1975. Member of A217 Department Heads Committee (steering committee and secretary), PM47 Tractor Committee (program chairman and chairman), former member of PM59 Greenhouse Mechanization, PM57 Controlled Traffic, PM43 Machinery Management (secretary, program chairman and chairman), PM44 Forage Committee (secretary, program chairman), and EPP357 Vegetable Oil-Diesel Fuel Alternative.

Member of NSPE and Montana Society of Engineers

Charter member and Past President of the Bozeman Chapter of the Montana Society of Engineers and past state president.

National Society of Professional Engineers-National Director, Governor for Montana PEE div., Western Region Vice Chairman for NSPE-PEE.

Member of Bozeman Engineers Club

Member of ASEE

Member of Sigma XI

Member of CAST

Other Assigned Duties:

Departmental Administration - Administrator of a department of eight faculty and approximately 100 students. The students represent two curriculums, Agricultural Engineering and Mechanized Agriculture. Faculty have split appointments with the College of Engineering, the Agricultural Experiment Station and the Cooperative Extension Service.

Teaching one to two courses on Advanced Design and Machinery per year.

The department head also has a research project in the Agricultural Experiment Station on the development of alternative grain drill openers for conservation farming plus grants and contracts research in alternative logging systems and no-till plot drill design.

Professional Activities

ASAE Department Heads Meetings
NSPE-PEE serving second term as the Western Region vic-chairman for the Professional Engineers in Education.
Responsibilities include planning for national meetings, developing education goals for NSPE and serving as chairman of the Faculty Affairs Committee of PEE.

Coop Faculty

The faculty member has served as the coordinator and recruiter for a coop program with the Soil Conservation Service.

W I L L I A M R . T A Y L O R , P h . D . , P . E.

EDUCATION

B.S. In Industrial Engineering, Oklahoma State University, 1963.
M.S. In Industrial Engineering, University of Arkansas, 1967.
Ph.D. In Industrial Engineering, University of Arkansas, 1969.
Major: Industrial Engineering Minor: Applied Statistics

PROFESSIONAL EXPERIENCE

Professor, Department of Industrial and Management Engineering, Montana State University, Bozeman, Montana, 1981-present.

Academic appointment with related responsibilities. Taught two courses per quarter when involved in research; otherwise three courses per quarter. Had several sponsored research projects ($165,000 in 1981-82) related to productivity in the timber industry. Served on a number of department and college committees. Held several positions in professional societies. Acted for Department Head in absence of same. Served on six to eight graduate committees per quarter. Advised approximately twenty-five undergraduates each quarter.

Quality Assurance Manager, Time Products Division, Texas Instruments, Inc., Lubbock, Texas, 1979-80.

Administered an annual budget of $500,000 with responsibilities for defining and controlling production processes and materials to assure product conformance to product drawings and quality goals. Responsibility included planning and forecasting of budgets, scrap, waste, cost reductions, and personnel, as well as, the development of Tactical Action Plans for achieving product quality and cost goals. Daily exercised judgment concerning the shipment of products that were not 100% defect free by relating them to customer satisfaction, product liability, and product billing plans. Supervised a staff of twenty-five engineers, technicians, and inspectors.

Reliability Engineer, Time Products Division, Texas Instruments, Inc., Lubbock, Texas, 1978-79.

Responsible for component and wrist watch qualification, environmental testing and failure analysis, design review and evaluation, management of all division reliability programs. Statistically forecasted product life, failure rates, and warranty returns. Supervised several engineering technicians. Made several trips to Asia to solve vendor related problems.

Associate Professor, Department of Industrial and Management Engineering, Montana State University, Bozeman, Montana, 1973-78.

Academic appointment with related responsibilities. Taught two courses per quarter when involved in research; otherwise three courses per quarter. Principal Investigator for several research projects ($102,210) related to productivity in Montana Hospitals. Served on a number of graduate, department, and university committees. Advised approximately twenty students per quarter.

Assistant Professor, Department of Industrial Engineering, Montana State University, Bozeman, Montana, 1968-73.

Academic appointment with related responsibilities. Taught three courses per quarter. Developed several undergraduate and graduate courses. Advised approximately twenty students per quarter. Conducted sponsored research ($45,000) related to timber harvesting systems. Served as Faculty Adviser to the student chapter of IIE.

National Science Foundation Trainee, Department of Industrial Engineering, University of Arkansas, Fayetteville, Arkansas, 1966-68.

Industrial Engineer, Southwestern Bell Telephone Company, Oklahoma City, Oklahoma, 1963, 1965.

Responsible for inventory record system of company owned but jointly-used service facilities. Conducted statistical analyses of data for use in contract negotiations.

PUBLICATIONS AND PRESENTATIONS

"Quality Assurance--A Survival Plan for the Next Decade", presented to the Eastern Idaho Section of the American Society of Quality Control (ASQC), Idaho Falls, Idaho, April, 1985.

"A Cable Forwarding System--Test Facility Design", William E. Larsen, Michael J. Gonsior, and William R. Taylor, Proceedings of the National Conference of the American Society of Agricultural Engineers, Paper No.84-1616, New Orleans, Louisiana, December 1984.

"Bunching Timber With a Radio-Controlled Winch In Mountainous Terrain," David F. Gibson, William R. Taylor, and Michael J. Gonsior, American Society of Agricultural Engineers Transactions, Vol. 27, No. 5, July-August, 1984.

"A Mathematical Programming Model for Assembling Unit Loads of Timber," William R. Taylor and Christopher B. Lofgren, Proceedings of the 34th Annual IIE Conference and Convention, Chicago, Illinois, May 1984.

"Fuzzy Sets--Computer Applications for Industrial Engineers," Edward L. Mooney, William R. Taylor, and Mark Amunrud, Proceedings of the 34th Annual IIE Conference and Convention, Chicago, Illinois, May 1984.

"Simulation Model for Assessing Harvesting and Handling Systems," William R. Taylor and David F. Gibson, Proceedings of the National Conference of the American Society of Agricultural Engineers, Chicago, Illinois, December 1983.

"Pre-Bunching Small Timber in Steep Terrain," William R. Taylor and David F. Gibson, Paper No. 82-1581, Proceedings of the National Conference of the American Society of Agricultural Engineers, Chicago, Illinois, December 1982.

"Industrial Engineering and Quality - A Broadening Responsibility," William R. Taylor, IE & Methods Management Bulletin, Number 220, Prentice Hall, Waterford, Connecticut, October 25, 1981.

"Quality Assurance In New Products Via Comprehensive Systems Approach," William R. Taylor, Industrial Engineering, Vol. 13, No. 3, March 1981.

PROFESSIONAL AFFILIATIONS

Registered Professional Engineer (Montana, Washington)
Institute of Industrial Engineers (IIE)
President, IIE, Yellowstone Chapter, 1981-83
Secretary-Treasurer, IIE, Yellowstone Chapter, 1977-79
Newsletter Editor, IIE, Engineering Economy Division
Student Chapter Adviser, IIE, 1969,72-75,80-present
Program Chairman, Engineering Economy Division, IIE National Conference and Convention, 1977
American Society of Engineering Economy (ASEE)
Secretary-Treasurer, ASEE Engineering Economy Division, 1972
American Society of Quality Control (ASQC)
Material Handling Institute Speaker's Bureau
National Science Foundation Traineeship Recipient
Reviewer, The Engineering Economist, 1983-
Alpha Pi Mu (National Industrial Engineering Honor Society)
Faculty Adviser, Alpha Pi Mu
Sigma Tau (National Engineering Honor Society)
Most Influential IE Professor (Student Award via Alumni Foundation), 1983, 1984
Halliburton Award for Teaching Excellence, 1984
Chairman, IIE Region X, Education & Professional Development
Ralph R. Teetor Educational Award, Society of Automotive Engineers, 1985

PERSONAL DATA

Date of Birth: October 25, 1939
Place of Birth: Borger, Texas
Height: 6 feet Weight: 170 pounds
Marital Status: Married, 2 Daughters
Health: Excellent

Compiled May, 1985

RESUME

Michael J. Gonsior

Education:

1962 Rensselaer Polytechnic Institute, Troy, NY
Bachelor of Science in Civil Engineering

1965 Montana State University, Bozeman, MT
Master of Science in Civil Engineering

Supplemental Academic Study:

1966 Operations Research; 80 hours, University of Michigan

1967 Finite Element Methods; 40 hours, Massachusetts Institute of Technology

1969 Soil Mechanics; 3 quarters, University of California

Professional Experience:

1962 to 1963 Research Assistant, Rensselaer Polytechnic Institute, Troy, New York.

1963 to 1965 Research Assistant, Montana State University, Bozeman, MT

1965 to 1972 Research Engineer, GS-7 to GS-11, RWU FS-INT-3701, Bozeman, MT

1972 to 1973 Research Engineer, GS-11, Falcon Project, PNW Station, Portland, Oregon.

1973 to present Research Engineering. GS-11 - GS-13. RWU FS-INT-3251, Missoula, MT.

Professional Activities and Recognition:

Honors and Awards:

Chi Epsilon, Tau Beta Pi 1962

Ideal Cement Company Fellowship, Montana State University - 1964-65.

Inventor's Award - U.S. Department of Commerce, April 16, 1984.

PUBLICATIONS LIST

Gonsior, M.J., James R. Ullrich, "Problems and opportunities in integrating and extrapolating research results," In: Environmental consequences of timber harvesting in rocky mountain coniferous forests: Proceedings of the symposium; 1979 September 11-13; Missoula, MT. Gen. Tech. Rep. INT-90, Ogden, UT, U.S. Department of Agriculture, Forest Service, Intermountain Forest and Range Experiment Station; 1980: 497-514.

Gonsior, Michael J. "A new concept for using the unusable." In: Land use allocation: Processes, people, politics, professionals: Proceedings, 1980 Society of American Foresters convention, October, 1980; Spokane, WA.

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Benson, Robert E., M.J. Gonsior, "Tree damage from skyline logging in a western larch/Douglas-fir stand," Res. Pap. INT-268. Ogden, UT; U.S. Department of Agriculture, Forest Service, Intermountain Forest and Range Experiment Station, 1981. 15 p.

Gibson, David F., William R. Taylor, Michael J. Gonsior, "Bunching timber with a radio-controlled winch in mountainous terrain." Transactions of the ASAE (Vol. 27, No. 5, pp. 1270-1276, 1984). Published by the American Society of Agricultural Engineers, St. Joseph, MI.

Larsen, W.E., M.J. Gonsior and W.R. Taylor, "A cable forwarding system - test facility design," ASAE Paper 84-1616, 1984.

Gonsior, Michael J., Leonard R. Johnson, "Harvesting systems for lodgepole pine forests," In: Lodgepole Pine - The species and its management: Symposium Proceedings, 1985. Published by Cooperative Extension, Washington State University.

Patents:

U.S. Patent # 4,103,784, Single Line, Traction Driven Running Skyline System, August 1, 1978.

U.S. DEPARTMENT OF AGRICULTURE
SCIENCE AND EDUCATION

PROPOSAL BUDGET

OMB Approved 0525-0001
Expires 12/31/86

ORGANIZATION AND ADDRESS	DURATION PROPOSED	S&E USE ONLY
Montana State University Office of Grants and Contracts Bozeman, MT 59717	Months: 40	Months:
PRINCIPAL INVESTIGATOR(S)/PROJECT DIRECTOR(S) William E. Larsen and W. Robert Taylor	FUNDS REQUESTED BY PROPOSER	FUNDS APPROVED BY S&E (If different)

	S&E FUNDED WORK MONTHS				
	Calendar	Academic	Summer		
A. Salaries and Wages					
1. No. of Senior Personnel					
a. 2 (Co)-PI(s)/PD(s)	33.5			$ 134,000	$
b. ___ Senior Associates					
2. No. of Other Personnel (Non-Faculty)					
a. ___ Research Associates-Postdoctorate					
b. ___ Other Professionals					
c. 2 Graduate Students				28,800	
d. 2 Pre-Baccalaureate Students				14,494	
e. 1 Secretarial-Clerical				3,000	
f. 2 Technical, Shop, and Other				60,000	
Total Salaries and Wages				240,294	
B. Fringe Benefits (If charged as Direct Costs)				44,206	
C. **Total Salaries, Wages, and Fringe Benefits** *(A plus B)*				284,500	
D. Nonexpendable Equipment *(Attach supporting data. List items and dollar amounts for each item.)* Not subject to indirect charge				75,000	
E. Materials and Supplies				3,000	
F. Travel					
1. Domestic *(Including Canada)*				4,000	
2. Foreign *(List destination and amount for each trip.)*					
G. Publication Costs/Page Charges				1,200	
H. Computer (ADPE) Costs cost share					
I. All Other Direct Costs *(Attach supporting data. List items and dollar amounts.* Details of subcontracts, including work statements and budget, should be explained in full in proposal.) Miscelaneous at $295 per month				11,800	
J. **Total Direct Costs** *(C through I)*				379,500	
K. Indirect Costs *(Specify rate(s) and base(s) for on/off campus activity. Where both are involved, identify itemized costs included in on/off campus bases.)* 39.5% except item O				120,278	
L. **Total Direct and Indirect Costs** *(J plus K)*				499,778	
M. **Less Residual Funds** *(If applicable)*				0	
N. **TOTAL AMOUNT of this REQUEST** *(L minus M)*				$ 499,778	$
O. **COST SHARING**	$ 162,500				

NOTE: Signatures required only for Revised Budget — *This is Revision No.* ___

NAME AND TITLE (type or print)	SIGNATURE	DATE
PRINCIPAL INVESTIGATOR/PROJECT DIRECTOR		
AUTHORIZED ORGANIZATIONAL REPRESENTATIVE		

(6) DETAILED BUDGET (Summary shown on the Proposal Budget form)

	A. Salaries and Wages	B. Fringe Benefits
Salaries for principal Investigators W. E. Larsen and W. R. Taylor. A total of 33.5 scientist months over the 40 month project at an average salary of $4,000 per month.	$134,000	$29,480 (22 %)
Graduate Research Assistants, two at $800 per month for 18 months each.	$ 28,800	$ 576 (2 %)
Pre-baccalaureate Student Labor	$ 14,494	$ 290 (2 %)
Secretarial	$ 3,000	$ 660 (22 %)
Technical shop labor, two one-half time technicians for a total of 40 man months over the 40 month project at an average salary of $1,500 per month.	$ 60,000	$13,200 (22 %)
Total	$240,294	$44,206

C. Total Salaries, Wages and Benefits $284,500

D. Nonexpendable Equipment, these are the components that go together to form the Proof of Concept Test Facility. This equipment is not subject to indirect charges. $ 75,000

The itemized items used to assemble the proof of concept test facility for the ENDLESS CABLE CONVEYOR SYSTEM FOR TIMBER HARVEST IN ROADLESS AREAS are listed below. These are estimated prices and bids will be required for firm prices.

15	Gear box drives @ $750	$11,250
16	Hydraulic motors @ $1,000	$16,000
15	Cable sheaves @ $110	$ 1,650
2	Hydraulic brakes @ $450	$ 900
6	Hydraulic cylinders @ $100	$ 600
7	Hydraulic metering valves @ $500	$ 3,500
10	Hydraulic pumps @ $1,500	$15,000
4000	Pounds of structural steel @ $0.75	$ 3,000
25	Cable segments @ $150	$ 3,750
25	Cable connectors @ $50	$ 1,250
12	Angle sensors @ $150	$ 1,800
10	Load sensors @ $500	$ 8,000
Miscellaneous hydraulic and electrical lines, valves, controls and fittings.		$ 8,300
Total as shown in D.		$75,000

E. Materials and supplies. (Computer paper, disks and miscellaneous supplies)	$ 3,000
F. Travel. (Professional meetings and local travel)	$ 4,000
G. Publication page charges.	$ 1,200
H. Computer Costs (An AT&T 6300 has been assigned to this project and the cost is shown as a cost share).	
I. All other direct costs, these items are estimated at $295 per month for operating costs not shown under other headings.	$ 11,800
J. Total Direct Costs.	$379,500
K. Indirect Costs, the indirect cost is based on a 39.5 percent rate on all budget items except (D) the nonexpendable equipment used to construct the test facility.	$120,278
L. Total Direct and Indirect Cost.	$499,778
M. Less Residual Funds	none
N. Total Amount of Request	$499,778
O. Cost Sharing, most of the cost sharing is in kind support provided by the USDA Forest Service Forestry Sciences Laboratory and consists of the following items.	
One AT&T 6300 microcomputer with maximum memory. This computer was purchased with grant funds from the previous forest reasearch project and is dedicated to this project.	$ 6,500
Heated laboratory space, 1,600 square feet provided by the USDA Forest Service Forestry Sciences Laboratory. Estimated value at $0.25 per square foot per month.	$ 16,000
Collaborative assistance provided by Michael J. Gonsior, USDA Forest Service Research Engineer, estimated value of $2,000 per month for 40 months.	$ 80,000
Collaborative assistance provided by Ron Babbitt, USDA Forest Service Eletronics Technician, estimated value of $1500 per month for 40 months.	$ 60,000
Total Cost Share (Estimated)	$162,500

OMB Approved 0525-0001
Expires 12/31/86

UNITED STATES DEPARTMENT OF AGRICULTURE
SCIENCE AND EDUCATION

SPECIAL CONSIDERATIONS

NOTE: Check appropriate statements. Supply additional information when necessary.

☒ This project does not involve human subjects.

☐ This project involves human subjects.

☐ It was approved by the Institutional Review Board on _____ *(date)*

☐ Is scheduled for review by the Institutional Review Board on _____ *(date)*

See DHHS regulations regarding the use of human subjects, appearing in Title 45, Code of Federal Regulations, Part 46, Subtitle A.

☐ This project does not involve recombinant DNA research.

☐ This project involves recombinant DNA research.

Supply appropriate documents as required by NIH *Guidelines for Research Involving Recombinant DNA Molecules* (48 FR 24556–24581 and subsequent revisions.)

ADDITIONAL INFORMATION

OMB Approved 0525-0001
Expires 12/31/86

UNITED STATES DEPARTMENT OF AGRICULTURE
SCIENCE AND EDUCATION

PROTECTION OF HUMAN SUBJECTS

STATEMENT OF POLICY – Safeguarding the rights and welfare of human subjects at risk in activities supported by Science and Education is the responsibility of the institution to which support is provided. In order to provide for the adequate discharge of this responsibility, USDA policy requires a formal assurance that appropriate committees in each institution will carry out both initial review of proposals, and continuing review of supported projects. The Department also requires certification of such reviews. Procedures which meet Department of Health and Human Services (DHHS) requirements will meet USDA requirements (Secretary's Memorandum No. 1755 and PL 93-348 as implemented by Part 46 of Title 45 of the Code of Federal Regulations, as amended (45CFR46)).

1. INSTITUTION
Montana State University
Office of Grants and Contracts
207 Montana Hall
Bozeman, MT 59717

2. TYPE ☒ ORIGINAL ☐ FOLLOW UP ☐ REVISION

3. PROJECT NUMBER *(If known)*

4. TITLE

An Endless Cable Conveyor System for Roadless Timber Harvest

5. PRINCIPAL INVESTIGATOR
William E. Larsen and W. Robert Taylor

6. *Check the following applicable statement:*

☒ A – This project does not include activities involving human subjects.

☐ B – This project includes activities involving human subjects but can in no way be considered at risk. _____ *(Initials)*
(If this statement is checked, the person signing this form must also initial in the space provided.)

☐ C – This project, which includes activities involving human subjects, is pending review by an institutional committee as provided by our assurance. Certification of completion of the review will be provided as soon as possible. *(Certification will be re-submitted on Form S&E-84, identified as "Follow up" in item 2, and checked in box "D")*

☐ D – This project includes activities involving human subjects. Our institutional committee reviewed and approved it on _____, in accordance with our assurance approved by S&E and/or DHHS. The project will be subject to continuing review as provided for in that assurance.

7. SIGNATURE OF AUTHORIZED ORGANIZATIONAL REPRESENTATIVE	TITLE	DATE
See Form S&E-661 (Aug 84)	Vice President for Research	

FOR OFFICE USE ONLY

UNITED STATES DEPARTMENT OF AGRICULTURE
SCIENCE AND EDUCATION

OMB Approved 0525-0001
Expires 12/31/86

CURRENT AND PENDING SUPPORT

INSTRUCTIONS: • Record information for the preceding period -- this will help S&E analyze shifts in research support. (Concurrent submission of a proposal to other organizations will *not* prejudice its review by S&E.) • All current research to which principal investigator(s) and other senior personnel have committed a portion of their time must be listed, whether or not salary for the person involved is included in the budgets of the various projects. • Provide analogous information for all proposed research which is being considered by, or which will be submitted in the near future to, other possible sponsors including other USDA programs.

NAME *(List P.I. No.1 first)*	CURRENT (✓)	PENDING (✓)	PROJECT NUMBER AND AGENCY	AMOUNT Code: T= Total A= Annual	EXPIRATION DATE OF GRANT	PERCENT OF EFFORT COMMITTED	TITLE OF PROJECT
William E. Larsen	x		655 MSU Grants and Contract. Funded by U.S.D.A. Forest Service, Forestry Sciences Laboratory, Funding shifted to U.S.D.A. Competitive Grants	$ 32,720	6/30/85	0.35	Cable Forwarding System (joint project with PI.2)
W. Robert Taylor	x		655 MSU Grants and Contracts. Funded by U.S.D.A. Forest Service, Forestry Sciences Laboratory, Funding shifted to U.S.D.A. Competitive Grants	16,000	6/30.85	0.33	Cable Forwarding System (Joint project with PI.1)

ADDITION TO THE PROJECT DESCRIPTION:

Title: AN ENDLESS CABLE CONEVYOR SYSTEM FOR TIMBER HARVEST IN ROADLESS AREAS

Principal Investigators: William E. Larsen and W. Robert Taylor

A copy of the letter from Michael J. Gonsior, principal collaborator on this project, confirming the USDA Forest Service's collaboration on this project. (Attached)

United States Department of Agriculture
Forest Service
Intermountain Station
Forestry Sciences Lab.
Montana State University
Bozeman, Montana 59717

Reply to 1580

Date May 28, 1985

Dr. William E. Larsen, Head
Agricultural Engineering Dept.
Montana State University
Bozeman, Montana 59717

Dear Bill:

This is to confirm our commitment to cooperate with you to the extent described in your proposal to the USDA Competitive Research Grants Program for Forest and Rangeland Renewable Resources for Fiscal Year 1985.

Since I first proposed the endless cable conveyor concept, it has been my hope that we (USDA Forest Service Intermountain Research Station) and you (Montana State University) could proceed cooperatively in its development, at least to the stage that someone in private industry would be able to take over confidently. My intent has been to fund much of this work through cooperative research agreements with Montana State University. Unfortunately, our budget does not allow us to do this, at least during this fiscal year; and we may not be able to provide much funding in future fiscal years either.

Actually, I have always preferred that Montana State University play the lead role in the research and development needed to prove the concept. By successfully competing for funds to carry on with this work, you will avoid many of the funding uncertainties that we in the Forest Service must contend with each fiscal year.

Good luck, and be assured that we will support you to the full extent of our capability.

Sincerely,

MICHAEL J. GONSIOR
Research Engineer

ATTACHMENT 2

BROCHURE

CABLE CONVEYOR CONCEPT - A

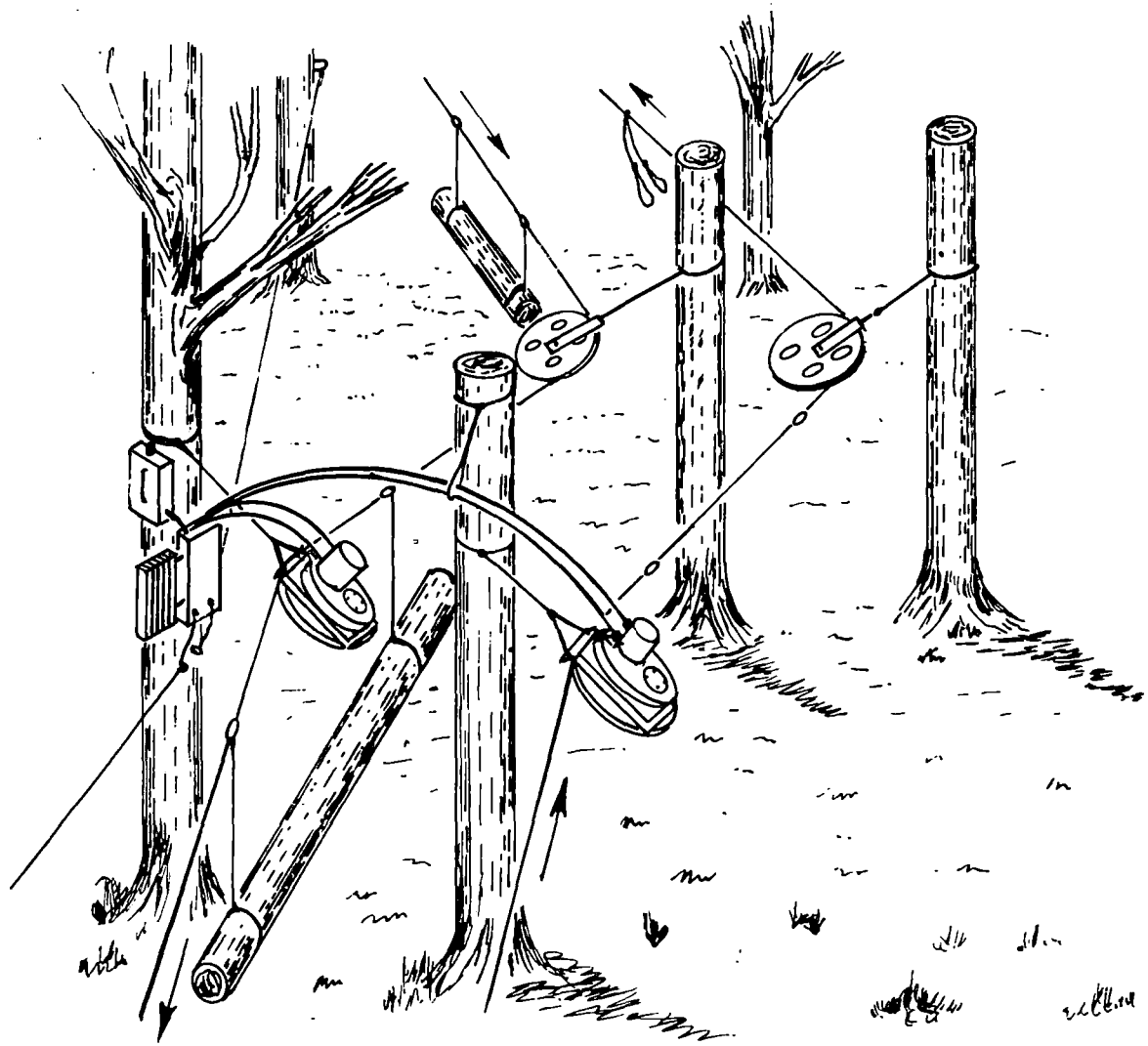
RESEARCH AND DEVELOPMENT PROSPECTUS

CABLE CONVEYOR CONCEPT
A RESEARCH AND DEVELOPMENT PROSPECTUS

USDA FOREST SERVICE
INTERMOUNTAIN RESEARCH STATION

AND

MONTANA STATE UNIVERSITY
COLLEGE OF ENGINEERING



AUGUST 1985

..without roadless harvesting alternatives the managers of many roadless forest areas may not be able to consider timber harvest as a management tool.

INTRODUCTION

Timber harvesting in the Interior West has been and continues to be controversial. In particular, the roads needed to enable timber harvesting with existing logging systems, especially in steep terrain, are objectionable to many people.

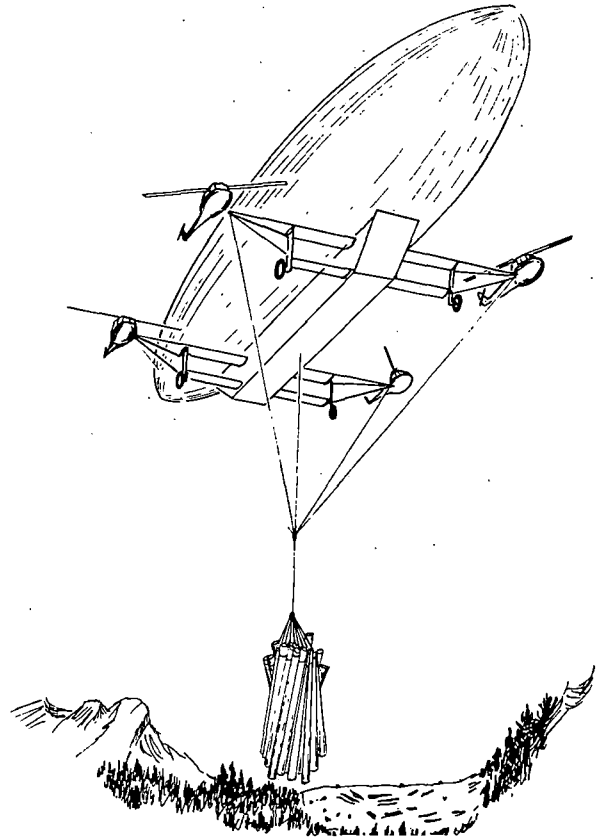
In response to the controversy surrounding road construction and logging in mountainous terrain, industry and government agencies have cooperated in recent decades to reduce the density of roads by using skyline systems, balloons, and helicopters. In addition, the environmental impacts of roads have been mitigated through more expensive construction methods. The result has been more costly logging, but seemingly without reduction of controversy. Indeed, some who object to the environmental consequences of logging in mountainous terrain are now complaining about the financial consequences as well.

No position is advocated or implied in this brochure regarding the merits of harvesting the timber in remaining roadless areas. It is merely recognized that the roads needed to enable such harvesting with existing logging systems may not be built in some areas, at least in the near future. Therefore, without roadless harvesting alternatives, the managers of such areas may not be able to consider timber harvest as a management tool.

The purpose of this brochure is to describe a concept that has been formulated to enable roadless timber harvesting in certain circumstances, and to outline the research and development work needed to test the feasibility of the concept. If this concept can be made to work, at least the controversy caused by logging roads can be dispelled, leaving only the merits of timber harvesting per se to be argued.

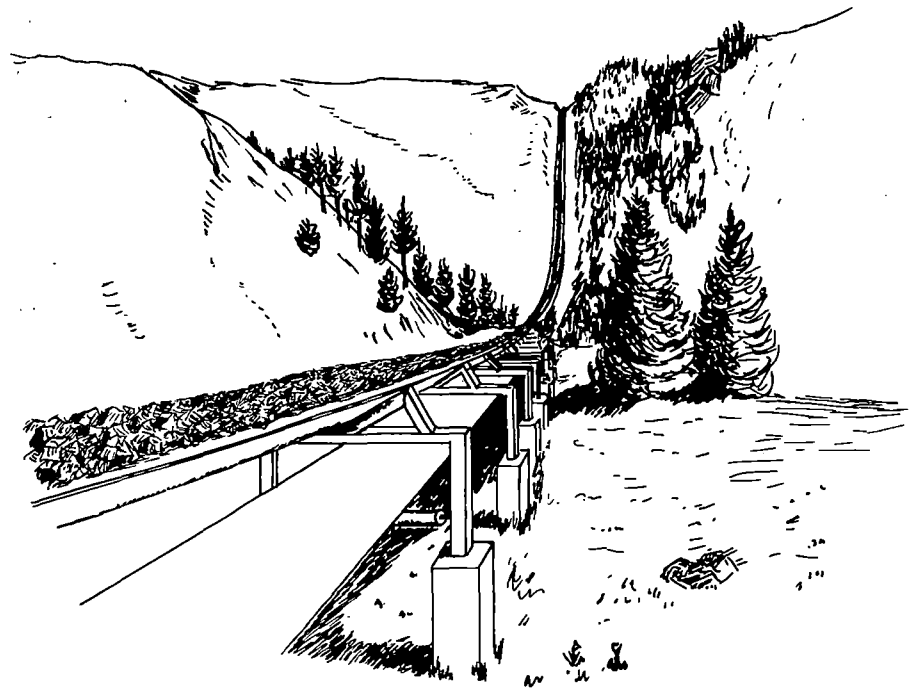
APPROACHES TO A SOLUTION

A few decades ago, short-reach cable systems requiring roads every few hundred feet were common throughout the Northern Rocky Mountains. As more and more objections to the environmental impacts of such road systems were raised, there was an evolution to skyline systems of greater and greater yarding distance capability. In addition, balloon-assisted cable systems as well as helicopters were introduced to reduce the need for roads. It is apparent that existing technologies are at the limits of their financial or technical feasibility. Indeed, there are now both private and government efforts underway to develop a new generation of heavy-lift aircraft to enable longer distance aerial yarding, presumably because the limits of existing helicopter technology have been reached. Relative merits of heavy-lift aircraft will be disscussed later in this brochure.



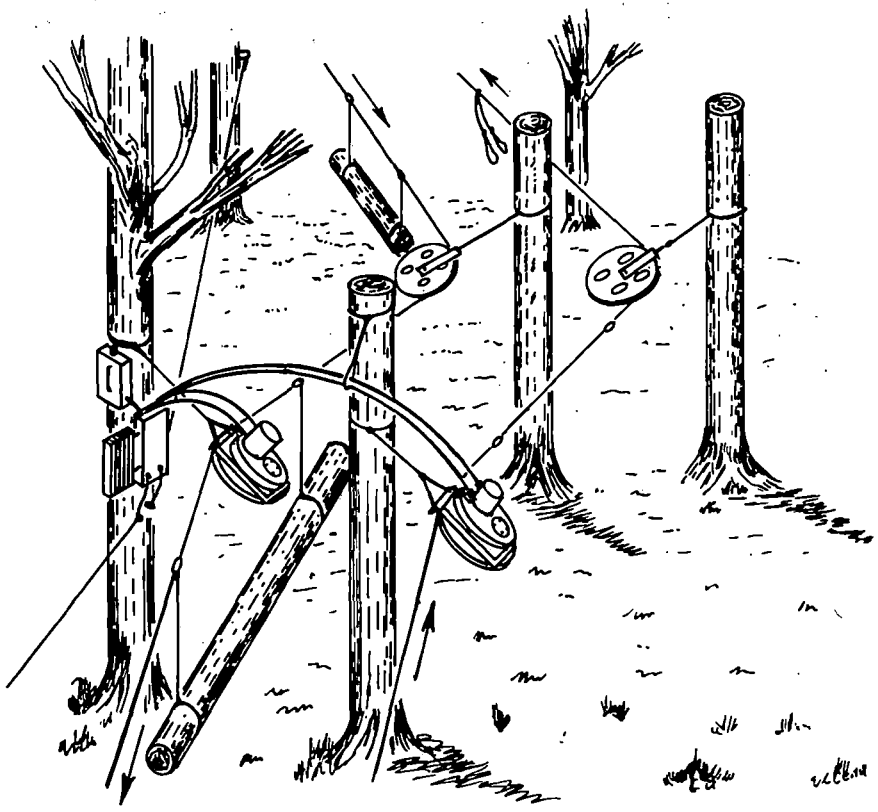
..efforts are underway to develop a new generation of heavy lift aircraft to enable longer distance aerial logging.

Common to nearly all contemporary systems for stump-to-landing transport, including heavy-lift aircraft, is that a vehicle or carriage travels one way loaded and one way unloaded. Consequently, production rates decrease in proportion to transport distance; and correspondingly, production costs increase. An alternative is continuous flow, the earliest example of which was probably the floating of logs in rivers. Other examples of continuous flow transport systems are flumes, conveyor belts, and pipelines.



..conveyor belts exemplify continuous flow transportation systems.

Preliminary analyses have lead to the conclusion that an endless cable conveyor is the most promising continuous flow system for woods-to-landing transport over long distances. Endless cable systems are used in a few circumstances in other parts of the world. The configuration known as the Japanese ZigZag System seems most promising, and it is the one upon which the concept described herein is based.

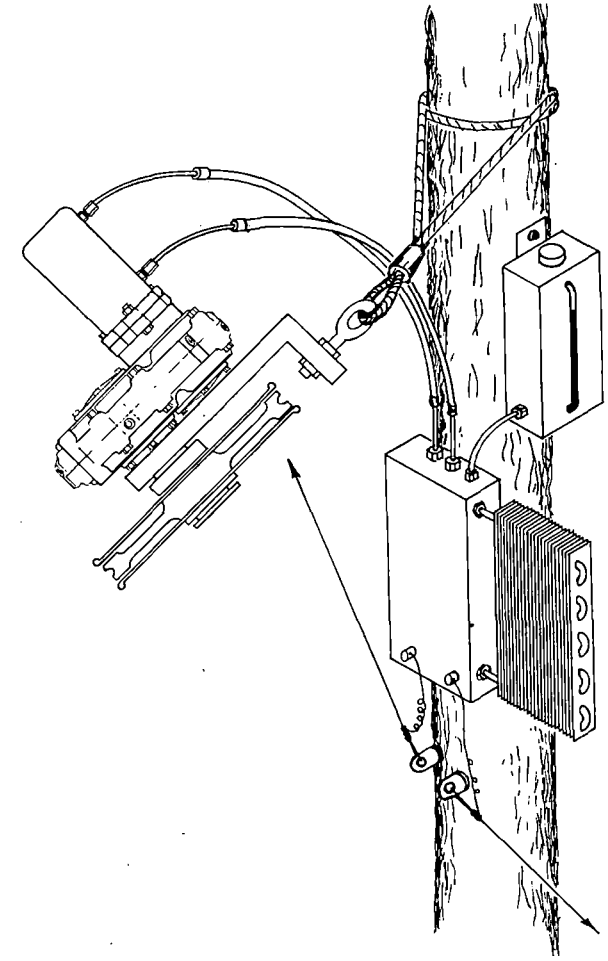


..an endless conveyor seems to be the most promising continuous flow system for woods-to-landing transport over long distances.

THE CABLE CONVEYOR CONCEPT

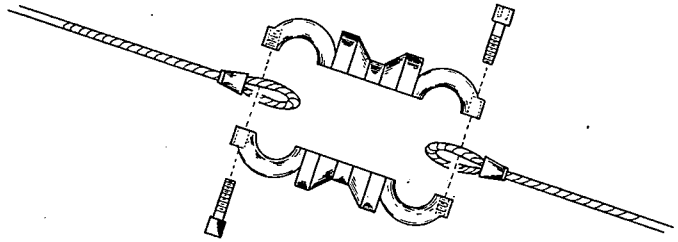
Existing endless cable systems, such as the Japanese ZigZag System, are readily adaptable to small timber. Generally the timber must be lifted and attached manually to the moving cable. Dismantling and moving such a system from one location to another often requires cable splicing to accommodate differences in route lengths. The length of route is limited, depending on size and frequency of loads as well as elevation difference between the upper and lower end of the system.

The research we have underway is to incorporate the endless cable concept in a system design that is not length or grade restricted. The concept employs control of cable tension and speed at intermediate locations along its route, analogous to intermediate pumping stations in a pipeline system or intermediate drive units along a conveyor belt. The control apparatus must be capable of monitoring cable tension and restraining the cable to avoid overload; and they must all be in communication with each other and with a master controller to maintain smooth operation and constant speed. Development of this cable speed and tension control is the major thrust of our ongoing and proposed research and development program.



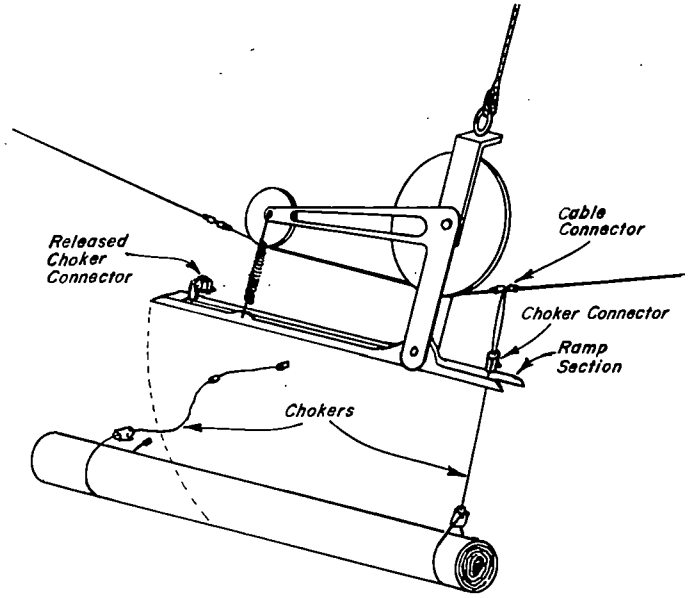
..the control apparatus must monitor and regulate cable tension and speed.

Another important element of our research is incorporation of quick connecting and disconnecting links between cable segments, to accommodate system length changes without the need for splicing. These links are also intended to serve as load attachment devices, to avoid cable abuse caused by concentrated loads.



..quick connecting and disconnecting links between cable segments will eliminate the need for cable splicing.

Finally, mechanisms for automatically loading and unloading the cable system must be developed, to enable safe and efficient loading of logs much larger than a worker can handle manually.

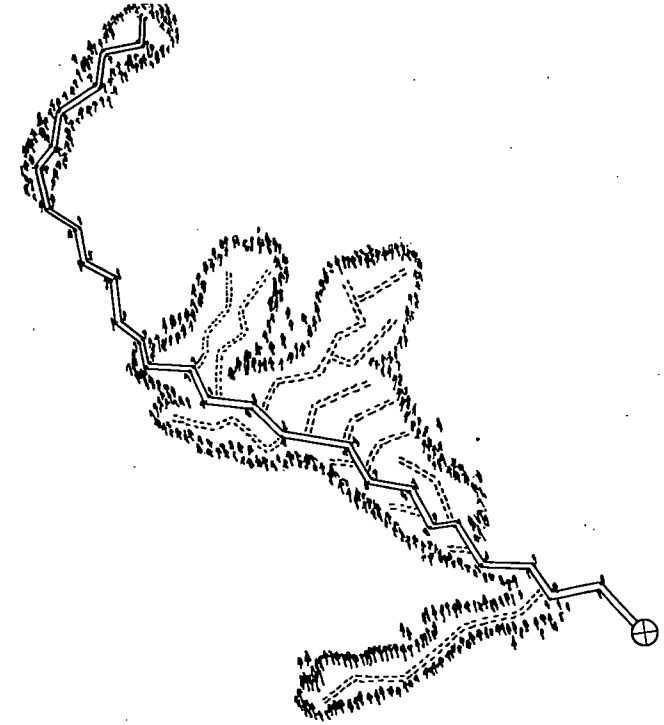


..the logs will be loaded and unloaded automatically.

SYSTEM DESIGN AND OPERATING CHARACTERISTICS

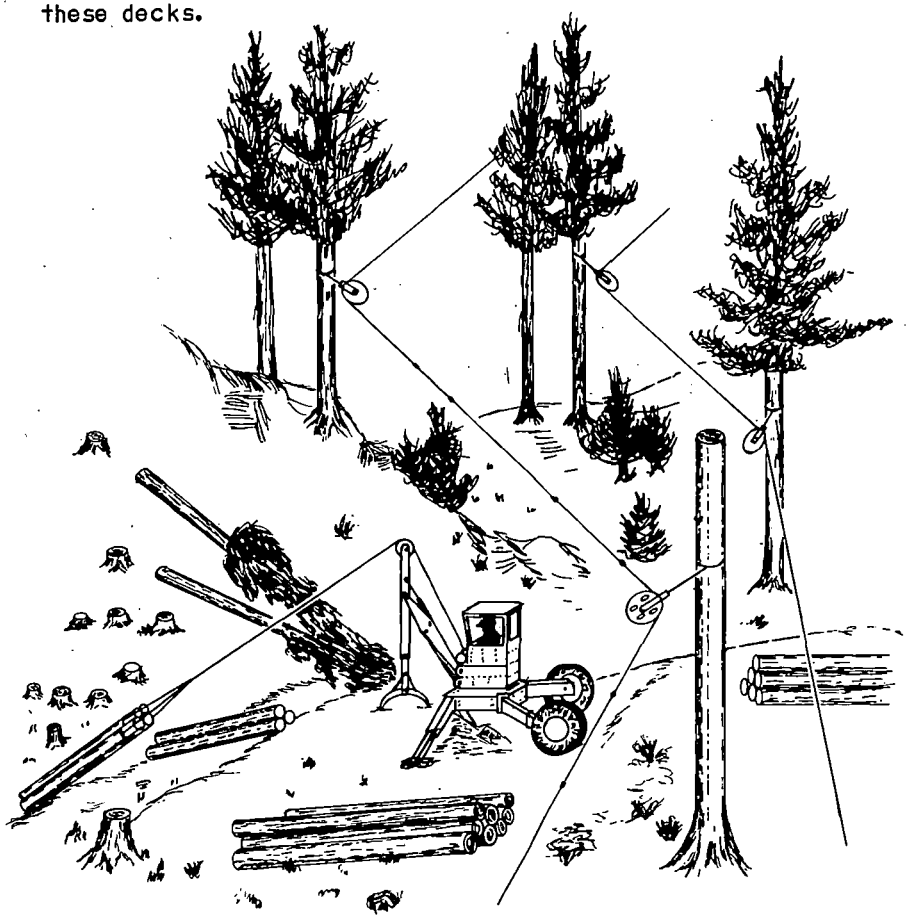
The Cable Conveyor System will be restricted primarily to downslope transport from woods to roadside landings, over routes of practically any length, grade, or alignment. Upslope transport of logs will be limited because of the difficulty of providing remote power along roadless routes. However, upslope transport of supplies, equipment and personnel is anticipated, utilizing some of the energy that must be dissipated in restraining the downward moving logs.

The proposed system will carry loads of 1/2 ton to 1 ton spaced about 500 feet apart at speeds up to 500 feet per minute. These loads are typical of tree sizes in many of the stand and terrain situations that appear to be likely candidates for such a system. This will result in a system capacity of 1/2 to 1 ton per minute when operated at full speed and load. A 3/4 inch diameter wire rope operating over spans of 500 feet or less should be well suited to this application.



..the system will be used primarily for downhill transport, probably following streamside routes, with excursions up tributary drainages and side slopes.

Streamside routes probably will be most favorable, with excursions up tributary drainages and side slopes to access the harvested timber. Where possible mechanized feller-bunchers will be used to cut the timber; but, whether cut by feller-bunchers or manually, the timber will need to be yarded and decked next to the planned cable conveyor route. Small cable yarders mounted on all-terrain vehicles may be used to make these decks.



..small scale yarders mounted on all-terrain vehicles may be used to deck the timber.

All-terrain vehicles may also be used for extension, dismantling, and moving of the conveyor system. The mechanism for retrieving logs and trees from the decks and loading them onto the moving cable will be mounted on an all-terrain vehicle. The log attaching machine will also serve as an off-loading station for chokers, fuel, and other supplies.

The logs and trees will be automatically off-loaded at the roadside landing. Conventional handling procedures will then be utilized depending on circumstances. The master control computer and cable tension control equipment will also be located at the landing. Chokers, fuel, and other supplies needed in the woods will be loaded onto the outbound segment of the cable conveyor at the landing.

COMPARATIVE ADVANTAGES OF THE CABLE CONVEYOR SYSTEM

The Cable Conveyor System can only be compared with aerial systems, since road-based systems are assumed to be unacceptable in the circumstances for which it is being proposed. Among aerial systems that can be compared are existing helicopters as well as proposed heavy-lift aircraft. However, presumably heavy-lift aircraft have already been determined superior to existing helicopters, as evinced by sustained support for their development. Therefore, the only comparisons necessary are between the Cable Conveyor System and heavy-lift aircraft.

Among the anticipated advantages of this type of system, compared with heavy- lift aircraft, are insensitivity to weather and lower hourly production rates. Delivery at the landing of only a few logs or trees at a time allows smaller landings with fewer workers and machines. One of the major problems with heavy-lift aircraft is the need for large, flat mooring areas for refueling and maintenance, which are usually rare in mountainous terrain.

Probably the major disadvantage of the Cable Conveyor System, compared with heavy-lift aircraft, is the need to bunch timber over longer distances in the woods. For heavy-lift craft, a load can be created almost anywhere in the woods. In clearcuts, loads for aerial systems might be created without need for any bunching, depending on tree or log sizes and tree densities. For the cable conveyor, logs and trees might have to be yarded over distances of up to 250 feet or more to optimize the overall logging cost. The cable conveyor is not nearly as portable as an aircraft; dismantling and moving it from one route to the next will be slow and relatively expensive. There will be a trade off between the cost of yarding the timber into decks in the woods and the cost of moving the Cable Conveyor System from one route to another.

Certainly the capital investment for the cable conveyor system will be lower than that for a heavy-lift aircraft. However, it must be acknowledged that heavy-lift aircraft can be used for a wide variety of tasks other than logging, which should enable them to be kept busy regardless of the health of the timber industry. On the other hand, even if heavy-lift

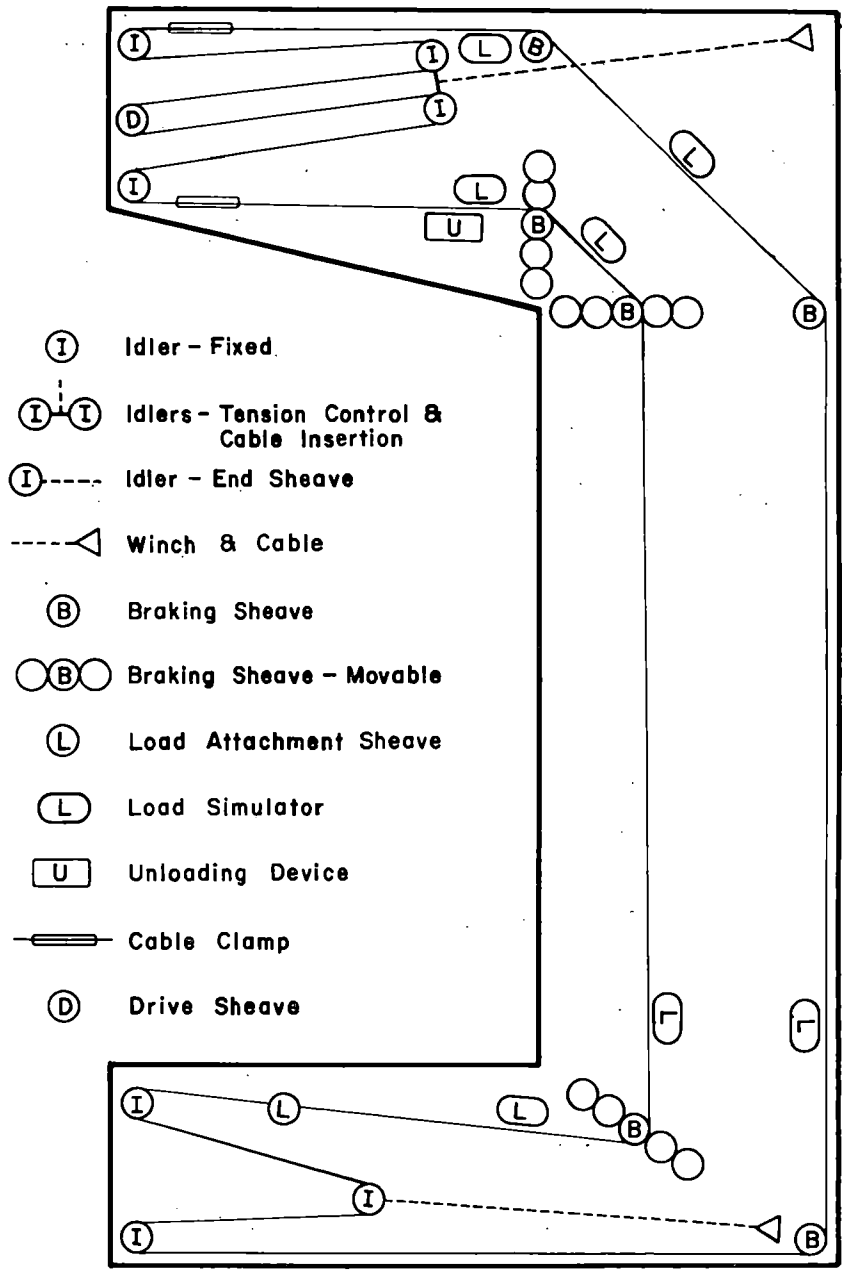
aircraft might be kept busy outside the logging industry, the large numbers of workers needed to prepare loads in the woods for them might not find it so easy to adapt to the "boom and bust" cycles so common in the timber industry. In this respect, the proposed Cable Conveyor Concept may be considered preferable, in that it requires fewer workers over a longer period of time to produce a given amount of timber.

Probably the most important comparison to be made is production cost, since neither heavy-lift aircraft nor the proposed Cable Conveyor System are expected to cause unacceptable environmental impacts. Preliminary analyses indicate that the Cable Conveyor System should be at least competitive with, and probably less expensive than, heavy-lift aircraft.

PROOF OF CONCEPT TEST APPARATUS

The next step in the research and development process is the fabrication and operation of a proof of concept test apparatus. Additional funding is required to build the proof of concept test apparatus and determine if the system will perform as anticipated. The computer controls and automatic loading and unloading systems are all concepts that must be tested and proven prior to field evaluation and demonstration.

The proof of concept test apparatus is shown here in a plan view. The apparatus will utilize half scale cables and operate at one-fourth the anticipated field loads. The proof of concept test apparatus will simulate field conditions for the evaluation of components. The load factor of one-fourth will permit scale up to field conditions with a high degree of confidence.



..proof of concept test apparatus will simulate field conditions for evaluation of components.

DEVELOPMENT SCHEDULE AND PROJECTED COSTS

The Cable Conveyor System appears to be a viable alternative for logging in roadless areas. Preliminary design and economic analysis have shown that the cable conveyor concept has sufficient merit to proceed with its development, at least through a stage needed to demonstrate its technical feasibility. The following diagram shows progress to date, the anticipated progression of steps to complete the research and development process, and the costs associated with each stage. The concept formulation and evaluation stages are complete and the preliminary design for a proof of concept test apparatus is nearing completion.

DEVELOPMENT SCHEDULE AND PROJECTED COSTS

PROJECT PHASE	TIME SCHEDULE, CALENDAR YEARS: COMPLETED 81	83	85	PROJECTED 87	89 . . ?	ESTIMATED COST, M$: PHASE	CUMULATIVE
PHASE I: CONCEPT	======					50	50
__Concept Formulation							
__Evaluation							
__Economic Analysis							
PHASE II: DESIGN		==========		=		90	140
__Component Design							
__Test Apparatus Design							
__Test Procedures							
PHASE III. TEST				==========		660	800
__Test System Fabrication							
__Proof of Concept Testing							
__Modification-Retest							
PHASE IV: TECHNOLOGY TRANSFER				==========?		?	?
__System Scale-up							
__Inwoods Testing							
__Transfer to Industry							

..concept formulation and evaluation stages are complete and the preliminary design for a proof of concept test apparatus is nearing completion.

AUTHORS

William E. Larsen, Professor of Agricultural Engineering, Michael J. Gonsior, Research Engineer, USDA Forestry Sciences Laboratory and W. Robert Taylor, Professor of Industrial Engineering, Montana State University, Bozeman, Montana.

ATTACHMENT 3

FLOW CHART AND COMPUTER PROGRAM FOR ANALYSIS OF MULTI-SPAN, MULTI-LOAD CABLE SYSTEM BETWEEN TWO CONTROL SHEAVES

One objective of the cooperative research investigation being reported herein was to develop a computerized analytic system to predict behavior of a prototype cable forwarding system, applicable as well to the proposed test facility. This Attachment and the following Attachment 4 contain the basic theory and algorithms for a crucial part of such a system--namely, the behavior of the cable and loads between two control sheaves. Unfortunately, lack of time prevented development of a complete simulation system incorporating many control sheaves, automated loading and unloading devices, and other features. Also needed are the means to predict behavior of a complete system during shutdown and start-up, as well as to simulate loosening (for extension or dismantling and moving) and subsequent retensioning and start-up.

The proposed system will consist of several control sheaves, at which the cable speed and tension will be monitored and controlled, and between which the cable may pass over several uncontrolled (freewheeling) sheaves. The cable itself is to be made up of segments, all of identical unstretched length, linked by specially designed connectors which also serve as load attachment devices. Thus, the unstretched length of cable between load attachment devices will be constant; but the stretched lengths between them will vary depending on the load magnitudes as well as their locations along the route.

Catenary theory is based on the assumption of inextensibility, which is acceptable for solution of many types of cable problems. However, for a very long cable system, with many loads distributed along its route, a more accurate theory that accounts for cable stretch seems desirable. Gonsior has derived such a theory; and a draft of his manuscript describing this theory is appended hereto as Attachment 4.

The flowchart and computer program contained in this attachment are designed to predict the status of the cable and loads in a system of spans between control sheaves. Relationships derived in Attachment 4 are used for certain computations.

Input requirements include the cable diameter and modulus (statements 120 and 140) from which the unstretched weight per unit length is found (statement 160). After the number of spans is specified (statement 170) the coordinates of each sheave--beginning with the uppermost control sheave and ending with the lowermost control sheave--are entered (statements 180 - 210)*. Next, the allowable tension and load spacing are entered (statements 215 and 220).

To estimate the number of loads (or load attachment devices) required for solution, the total of chord lengths in the system is determined (statements 250 - 280). This total length is divided by the load spacing, then increased by 2, and finally truncated to an integer (statement 290). The user may then specify identical loads (statements 300 - 340) or specify them otherwise (statements 350 - 420).

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*Note that the coordinates XS(I) and YS(I) have their highest values when I = 1, and they decrease progressively as I increases. (It is possible, of course, for YS to remain unchanged or even increase slightly as I increases from one intermediate integer value to the next; but overall, the elevation from the uppermost to the lowermost control sheave will decrease.) Note also that, even though a real system would follow a zigzag pattern, the XS values merely reflect the horizontal distances between the sheaves without regard for the bearings of the spans.

The first pass involves setting the cable tension at the uppermost control sheave equal to the allowable working strength (statement 460) as well as specifying the unstretched cable length from the uppermost control sheave to the first load (statement 470). Solution is initiated by assuming that the slope of the cable at the uppermost control sheave equals the chord slope of the first span (statement 520). The combination of this slope and the allowable working strength yields the horizontal component of tension in the span (statement 530). (The value of BK from statement 530 is the equivalent of H in Attachment 4, equations 10 and 14.) Solution then proceeds to find the slope of the cable on the upslope side of the first load immediately adjacent to the load (statement 620). (The value of T from statement 620 is the same as θ_0 in Attachment 4, equation 33.)

The horizontal and vertical distances from the uppermost control sheave to the first load are found from statements 630 - 640 and 650 - 660, respectively (which are the same as equations 21 and 22, respectively, in Attachment 4). A tentative value for the horizontal coordinate of the first load is then found (statement 670) and compared with the horizontal coordinate of the next sheave to determine if the first load is, indeed, within the first span (statement 680). If it is, then its vertical coordinate, YO, is found (statement 690). Also determined are the cable tension on the upslope side of the load (TA(K) from statement 720), the load magnitude (WT(K) from statement 740), and the cable slope and tension on the downslope side of the load immediately adjacent to the load (TH from statement 750 and TB(K) from statement 770, respectively). The index values J and K are then increased (statements 810 and 820) and the solution proceeds to the next load (back to statement 620).

If the comparison in statement 680 shows the load to be beyond the next sheave, then the horizontal distance from the previous load to that sheave is determined (statement 840) and, based thereon, an iterative scheme per equations 23, 23a, and 24 of Attachment 4 is used to find the slope of the cable on the upslope side of and adjacent to the sheave, AO(I), (statements 850 - 920). Thereafter the vertical elevation difference between the previous load and the next sheave is found, using equation 22 in Attachment 4 (statements 940 - 950), resulting in an estimate for the elevation of the sheave (statement 960). If this elevation estimate is more than a tolerable distance above or below the true elevation of the sheave (statement 970) a linear approximation scheme is used to establish a new value for the cable slope, TH, at the upper end of the span (statements 990 - 1030 the first time through, then statements 1070 - 1130 for all subsequent iterations), and the solution scheme is repeated (via statement 1050). This scheme is repeated until the calculated elevation of the next sheave is sufficiently close to its true elevation (per statement 970), after which the cable tension on the upslope side of the sheave immediately adjoining it is found (A(I) from statement 1170).

Depending upon whether there were loads in the span or not (statement 1180) different routines are used to find the unstretched length of cable in the span just analyzed, TL(I), as well as to the first load in the next span, SS(I + 1)(statements 1190 - 1210 if no loads in span, otherwise statements 1230 - 1250). Then the number of loads in the span, N(I), are retained in memory for later use (statement 1260) and the cumulative unstretched length of cable thus far is determined (statement 1270).

From statement 1280, it is determined if analysis of the entire system has been completed. If not, I is increased (statement 1290) and the tension at the uppermost extremity of the next span, B(I), is set equal to the tension at the lowermost extremity of the span just analyzed, A(I - 1)(statement 1300). The load index is then initialized (statement 1310) and solution continues at statement 510.

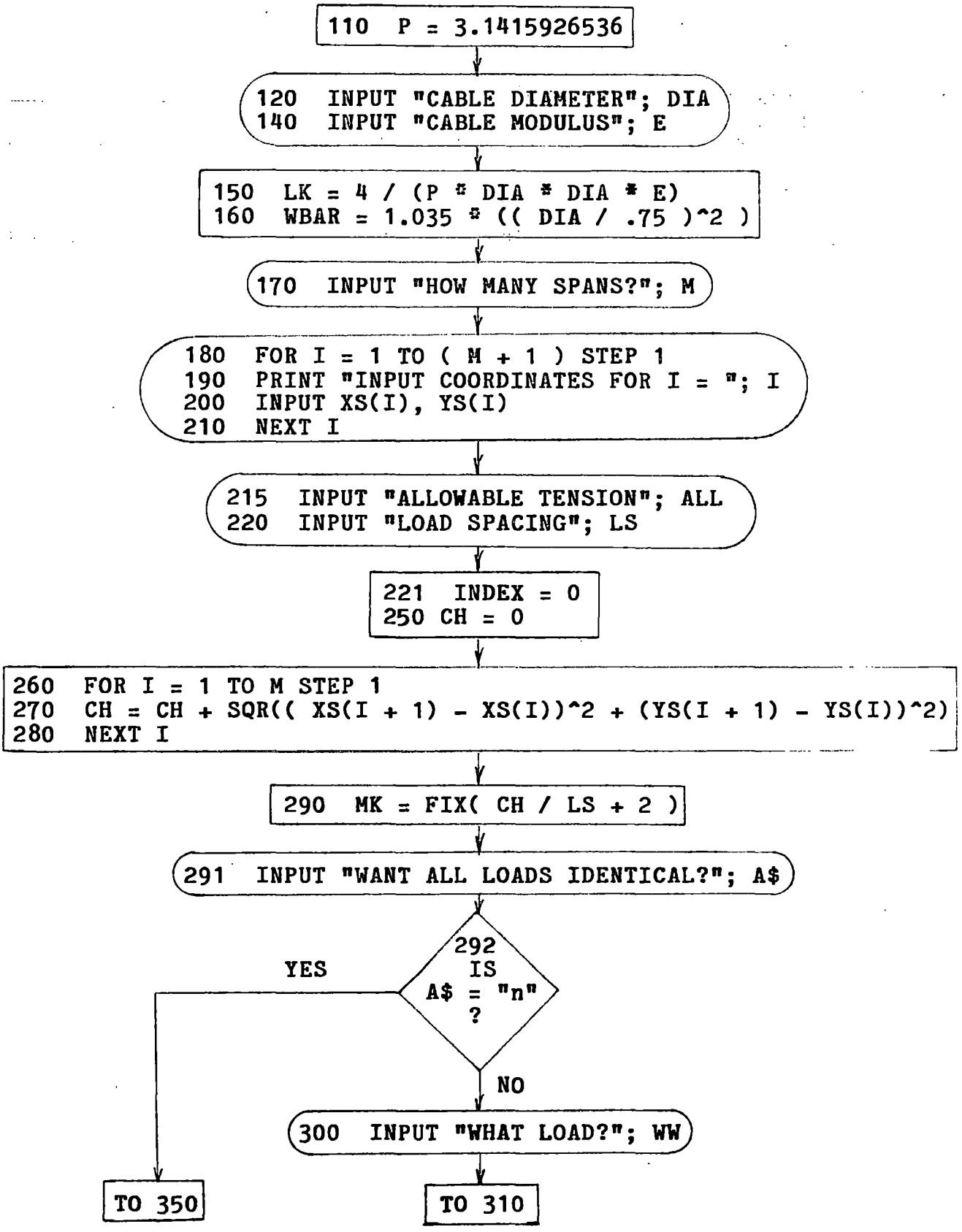
If it is determined at statement 1280 that solution has been completed, and if the solution mode is still "PASS 1" (statement 1330), then the total unstretched cable length just determined, LL, is compared with LMAX (statement 1340). Depending on the outcome of this comparison, the longest value is retained as LMAX and the corresponding distance from the uppermost control sheave to the first load, SC, is also retained (statements 1350 and 1360). Thereafter, the user may obtain an interim printout (statements 1520 - 1675 via statements 1370 - 1380) and/or specify another set of load positions (via statements 1390 - 1400). Otherwise, if the user is satisfied that enough load position sets have been analyzed, he may terminate PASS 1 (statement 1490) and enter "PASS 2".

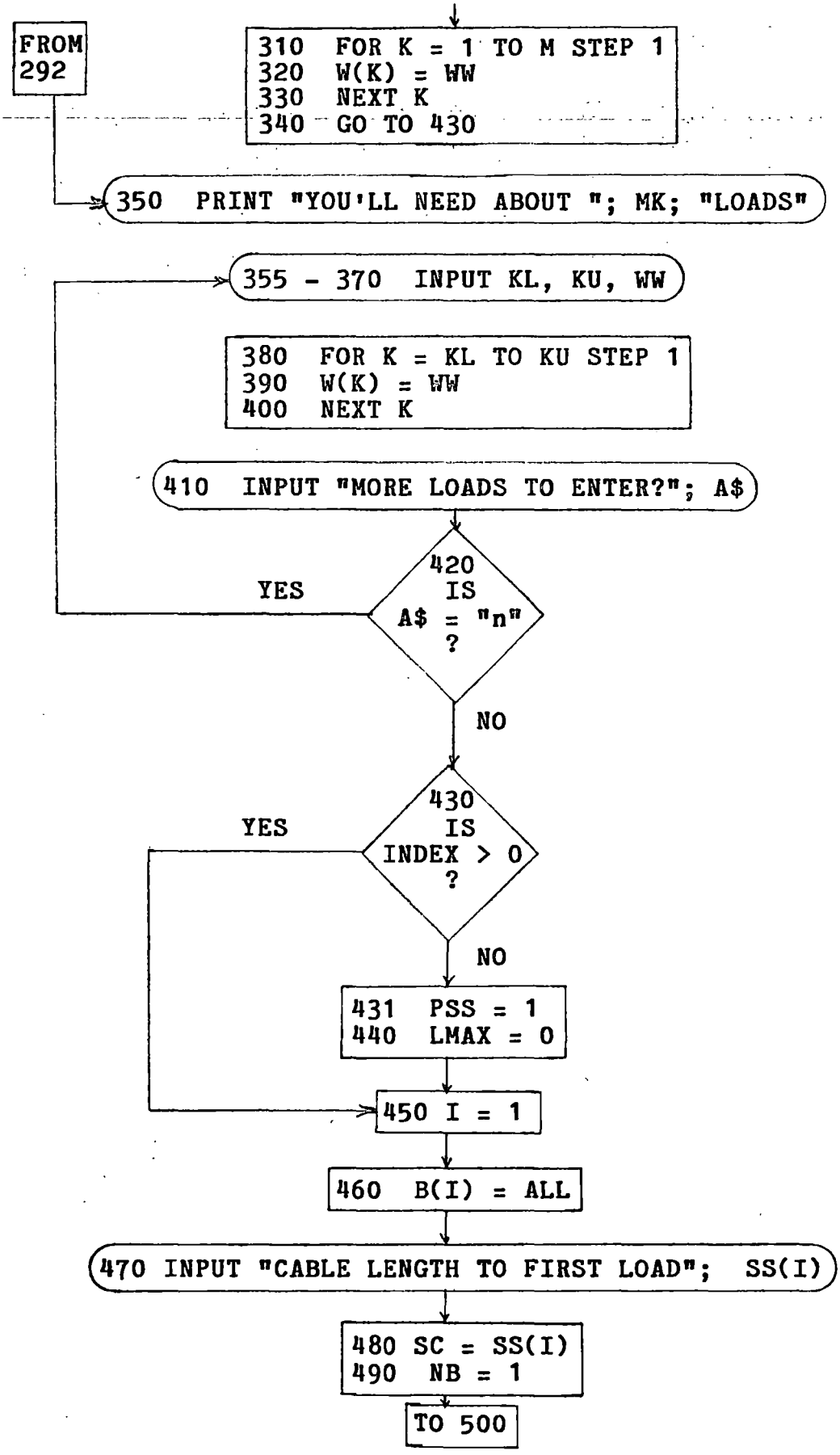
In PASS 2 the algorithms employed in PASS 1 are used, but from statement 1330 the program goes to statement 1730 where it is determined if the value of LL just computed is acceptably close to LMAX (the longest unstretched length found in PASS 1). If not, a linear approximation scheme is entered to adjust the value of B(1)(statements 1750 - 1810 the first time through, then statements 1830 - 1892 in subsequent iterations) so that, eventually, a solution is found that satisfies the criterion implied in statement 1730. Once satisfied, the predicted status of the cable and loads is printed via statements 1520 - 1675. The user is then free to change the load positions (statements 1690 - 1702) or even change the loads per se (statements 1703 - 1709). If desired, the user may even change the cable length without having to go through PASS 1 again (statements 1710 - 1713) with the option of simultaneously changing the loads (statements 1714 - 1715). Of course, if desired, a similar result can be obtained by merely reverting to PASS 1 (via statements 1716 - 1717).

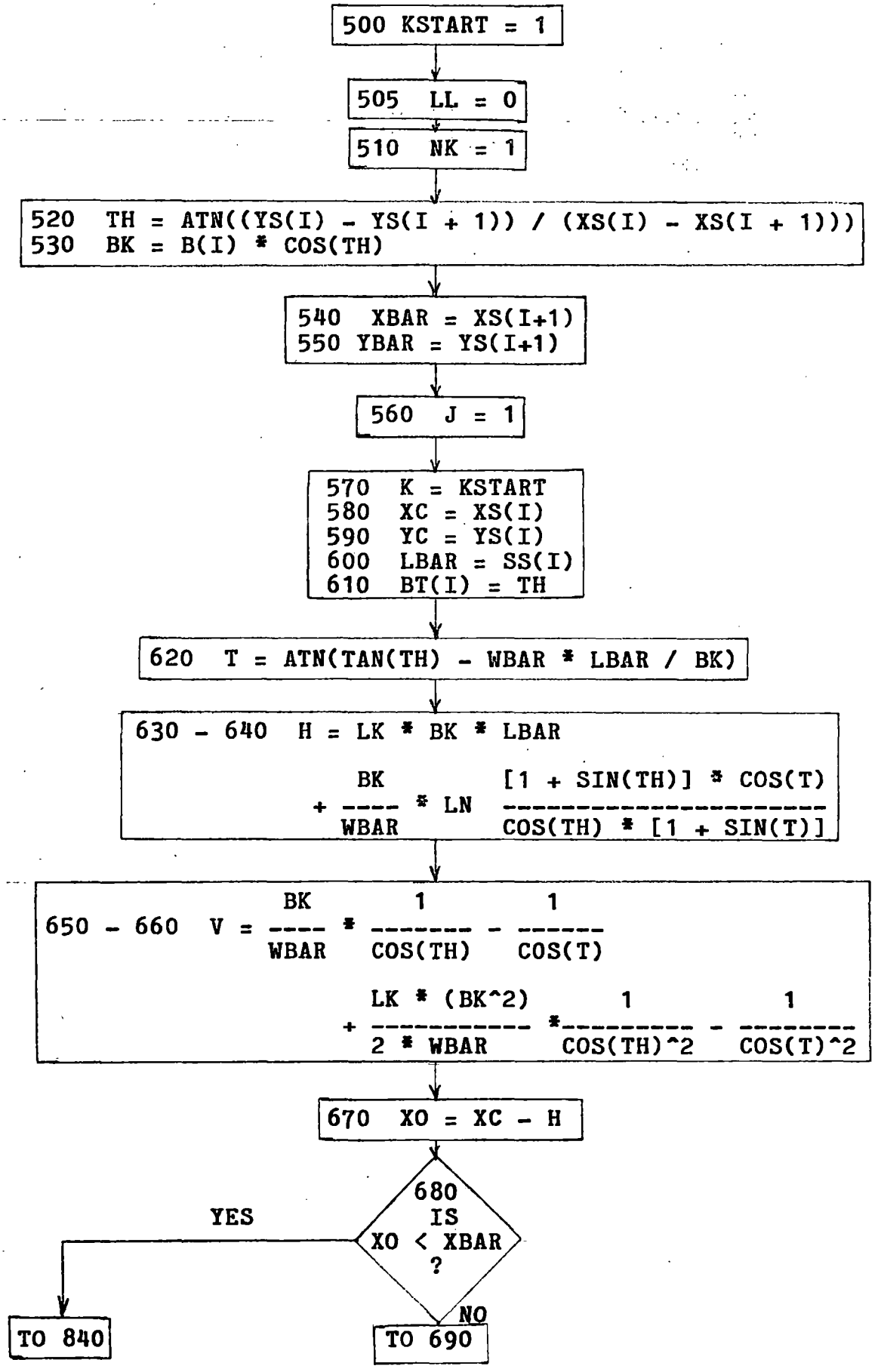
The reader is cautioned regarding the program listing contained herein, wherein the notation !(1) symbolizes double precision. Other symbols may be used, such as #, in which case alteration will be necessary. However, in so doing, be sure that the variable B!(1)--in statements 1760, 1790, 1810, 1860, 1891, and 1892--is retained as a subscripted variable. For example, B!(1) might become B#(1); whereas other variables such as LK!(1), WBAR!(1), etc. might become merely LK#, WBAR#, etc. Of course, variables subscripted with I, J, K, I + 1, etc. should retain their subscript notation in any transformation.

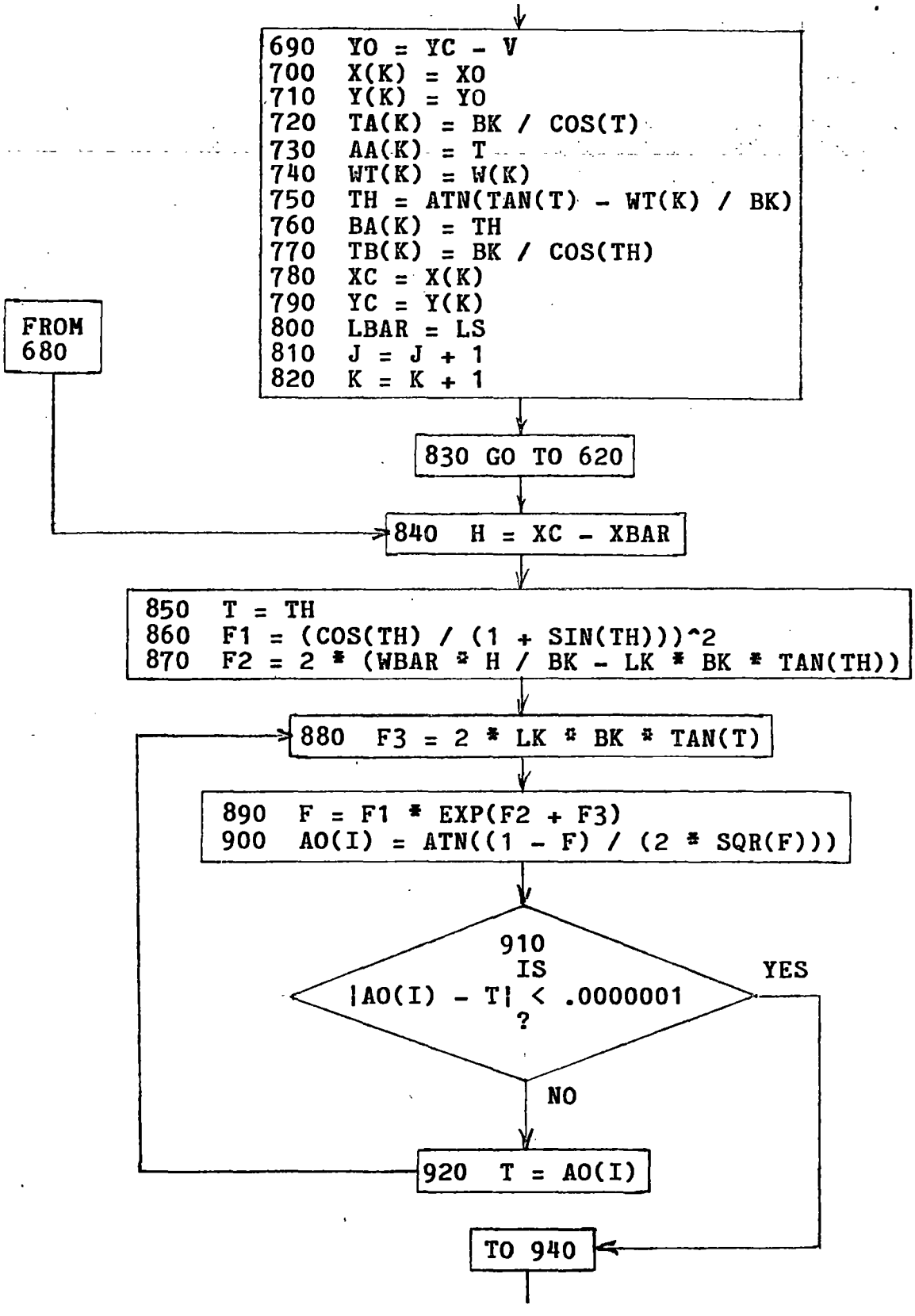
It should also be noted that the program has no dimension statements at its beginning. The system for which it was written requires no such statements provided subscript magnitudes are less than 10. For more general applicability the user may wish to add appropriate dimension statements.

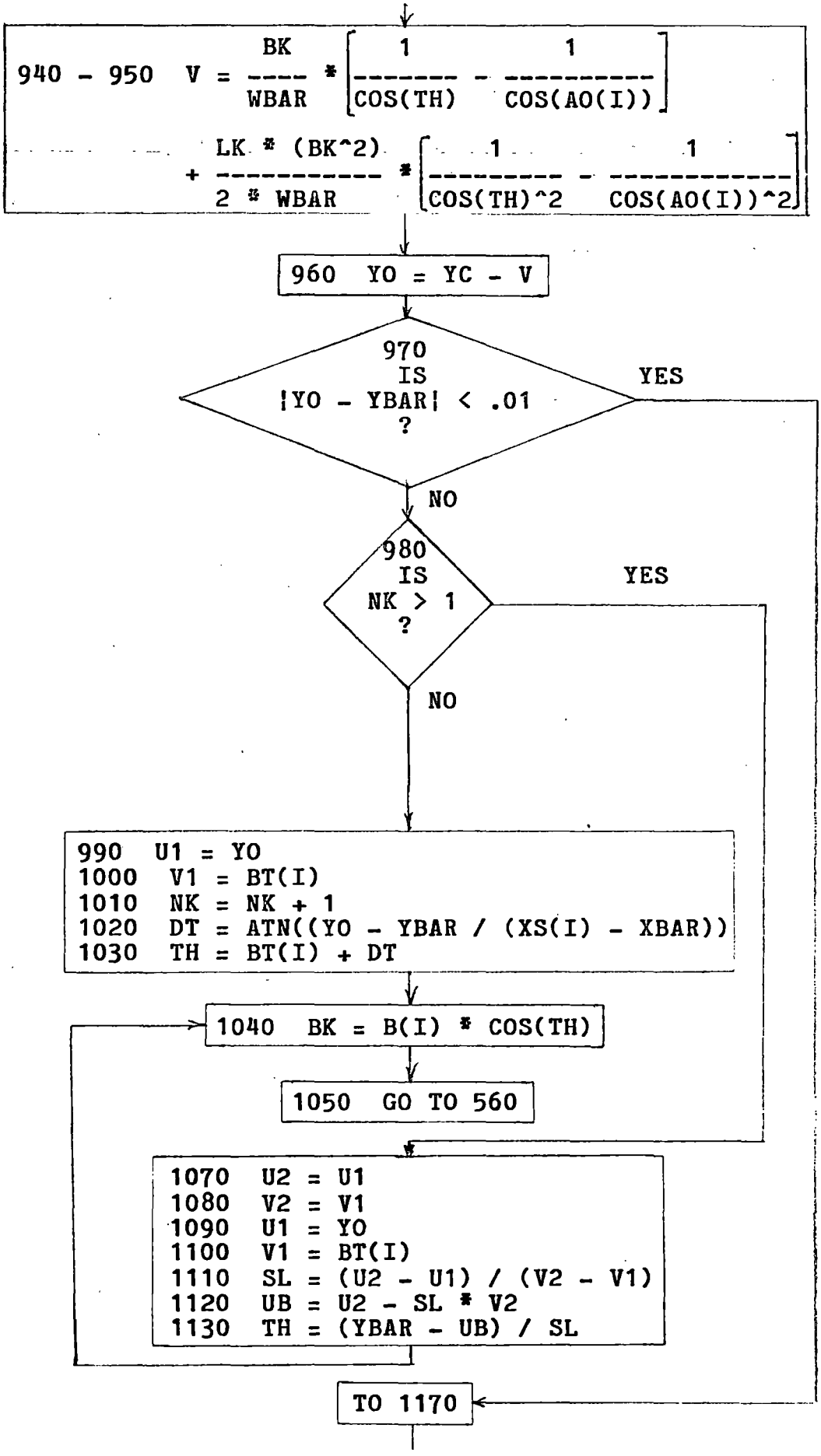
Other versions of this program have been written. (For example, the senior author of this report has written one that provides certain output in graphical format.) Such versions will not be included here, because the major purpose is to document the algorithms and logic in a manner that will enable future expansion to a complete system, comprising many control sheaves as well as loading and unloading mechanisms and other apparatus.

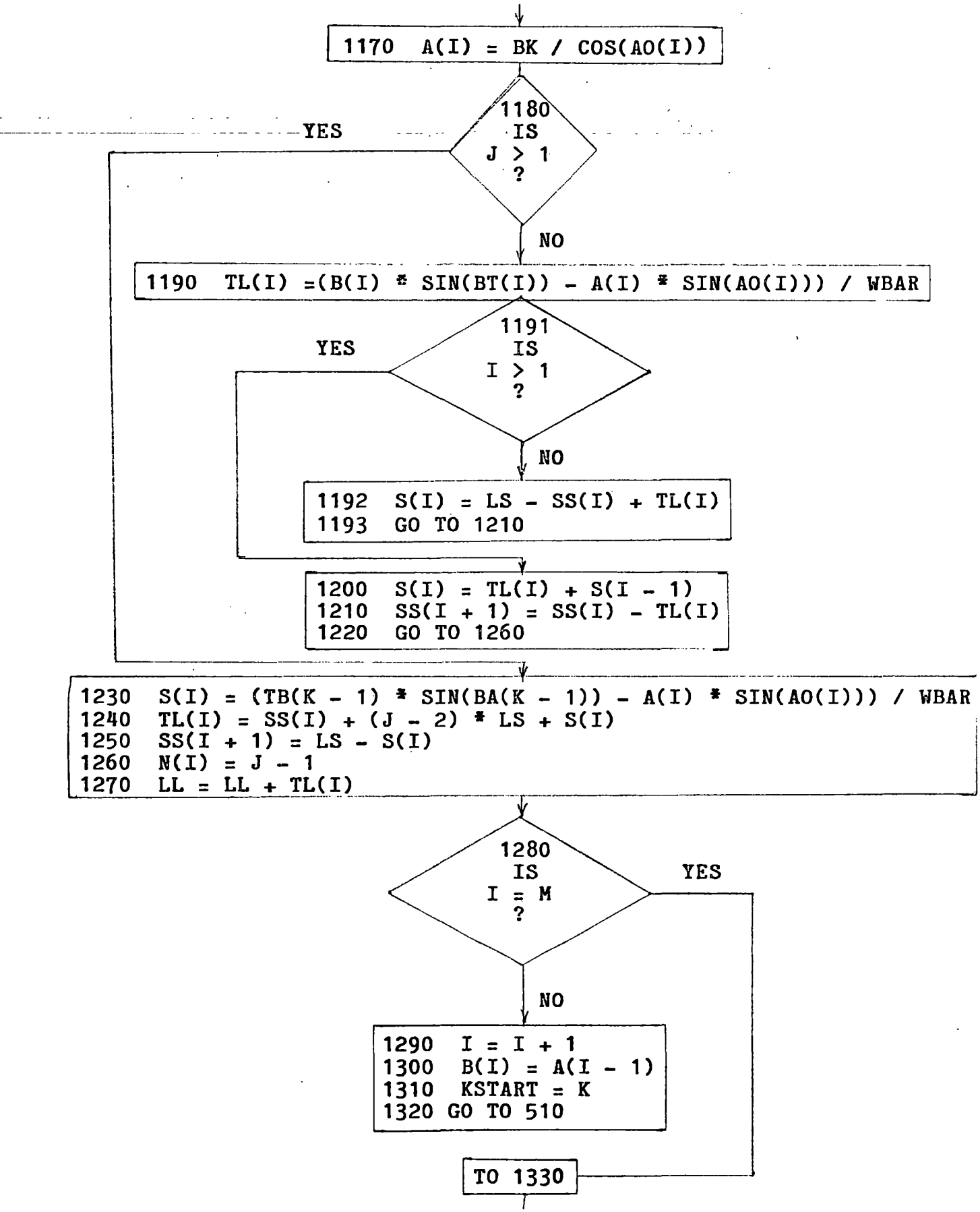

110 P = 3.1415926536
120 INPUT "CABLE DIAMETER"; DIA
140 INPUT "CABLE MODULUS"; E
150 LK = 4 / (P * DIA * DIA * E)
160 WBAR = 1.035 * ((DIA / .75)^2)
170 INPUT "HOW MANY SPANS?"; M
180 FOR I = 1 TO (M + 1) STEP 1
190 PRINT "INPUT COORDINATES FOR I = "; I
200 INPUT XS(I), YS(I)
210 NEXT I
215 INPUT "ALLOWABLE TENSION"; ALL
220 INPUT "LOAD SPACING"; LS
221 INDEX = 0
250 CH = 0
260 FOR I = 1 TO M STEP 1
270 CH = CH + SQR((XS(I + 1) - XS(I))^2 + (YS(I + 1) - YS(I))^2)
280 NEXT I
290 MK = FIX(CH / LS + 2)
291 INPUT "WANT ALL LOADS IDENTICAL?"; A$
292 IS A$ = "n" ?
YES
NO
300 INPUT "WHAT LOAD?"; WW
TO 350
TO 310

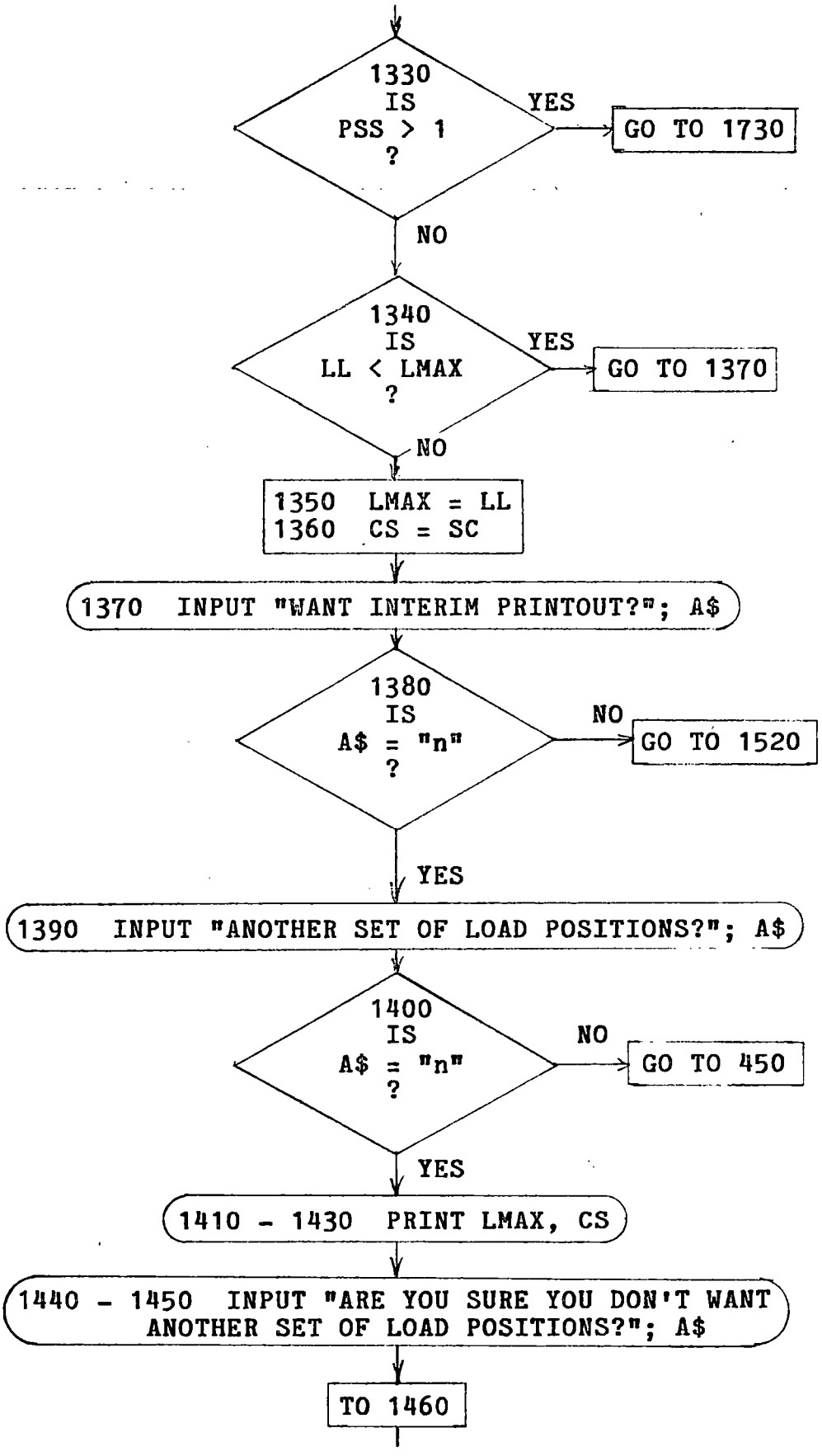

FROM 292
310 FOR K = 1 TO M STEP 1
320 W(K) = WW
330 NEXT K
340 GO TO 430
350 PRINT "YOU'LL NEED ABOUT "; MK; "LOADS"
355 - 370 INPUT KL, KU, WW
380 FOR K = KL TO KU STEP 1
390 W(K) = WW
400 NEXT K
410 INPUT "MORE LOADS TO ENTER?"; A$
420 IS A$ = "n" ?
YES
NO
430 IS INDEX > 0 ?
YES
NO
431 PSS = 1
440 LMAX = 0
450 I = 1
460 B(I) = ALL
470 INPUT "CABLE LENGTH TO FIRST LOAD"; SS(I)
480 SC = SS(I)
490 NB = 1
TO 500

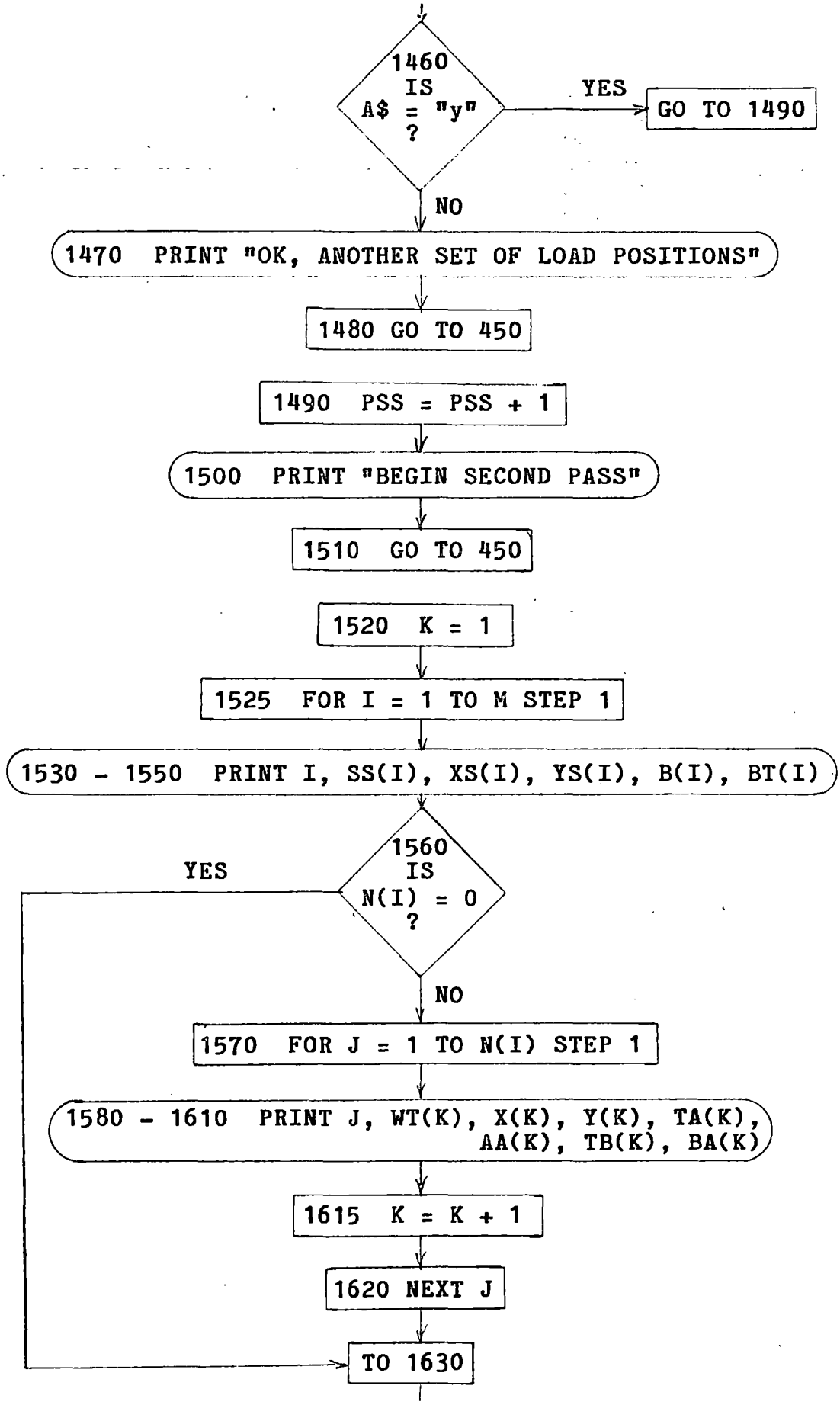

500 KSTART = 1
505 LL = 0
510 NK = 1
520 TH = ATN((YS(I) - YS(I + 1)) / (XS(I) - XS(I + 1)))
530 BK = B(I) * COS(TH)
540 XBAR = XS(I+1)
550 YBAR = YS(I+1)
560 J = 1
570 K = KSTART
580 XC = XS(I)
590 YC = YS(I)
600 LBAR = SS(I)
610 BT(I) = TH
620 T = ATN(TAN(TH) - WBAR * LBAR / BK)
630 - 640 H = LK * BK * LBAR + BK/WBAR * LN [1 + SIN(TH)] * COS(T) / COS(TH) * [1 + SIN(T)]
650 - 660 V = BK/WBAR * 1/COS(TH) - 1/COS(T) + LK * (BK^2)/(2 * WBAR) * 1/COS(TH)^2 - 1/COS(T)^2
670 XO = XC - H
680 IS XO < XBAR ?
YES
NO
TO 840
TO 690

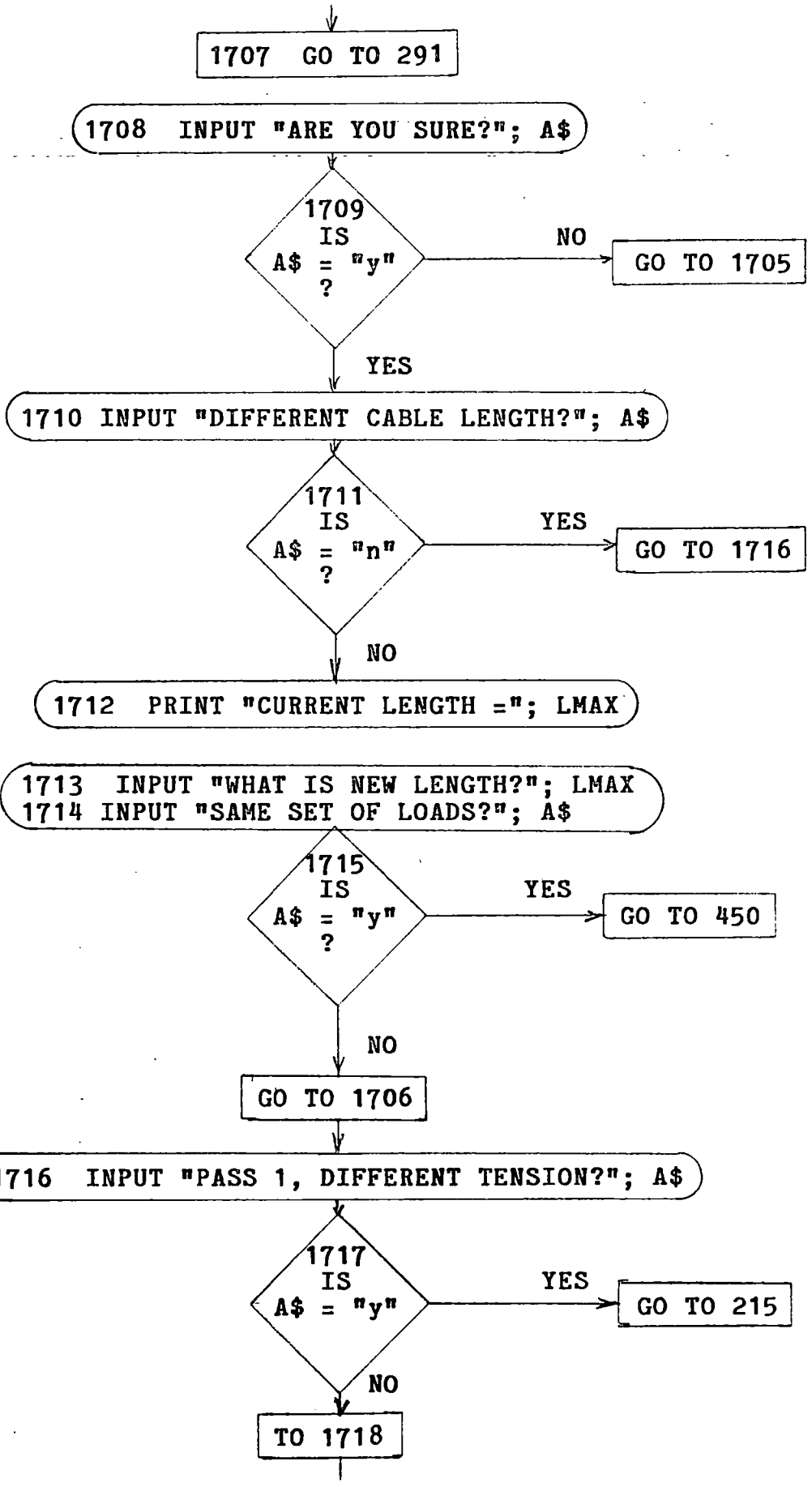

690 YO = YC - V
700 X(K) = XO
710 Y(K) = YO
720 TA(K) = BK / COS(T)
730 AA(K) = T
740 WT(K) = W(K)
750 TH = ATN(TAN(T) - WT(K) / BK)
760 BA(K) = TH
770 TB(K) = BK / COS(TH)
780 XC = X(K)
790 YC = Y(K)
800 LBAR = LS
810 J = J + 1
820 K = K + 1
FROM 680
830 GO TO 620
840 H = XC - XBAR
850 T = TH
860 F1 = (COS(TH) / (1 + SIN(TH)))^2
870 F2 = 2 * (WBAR * H / BK - LK * BK * TAN(TH))
880 F3 = 2 * LK * BK * TAN(T)
890 F = F1 * EXP(F2 + F3)
900 AO(I) = ATN((1 - F) / (2 * SQR(F)))
910 IS |AO(I) - T| < .0000001 ?
YES
NO
920 T = AO(I)
TO 940

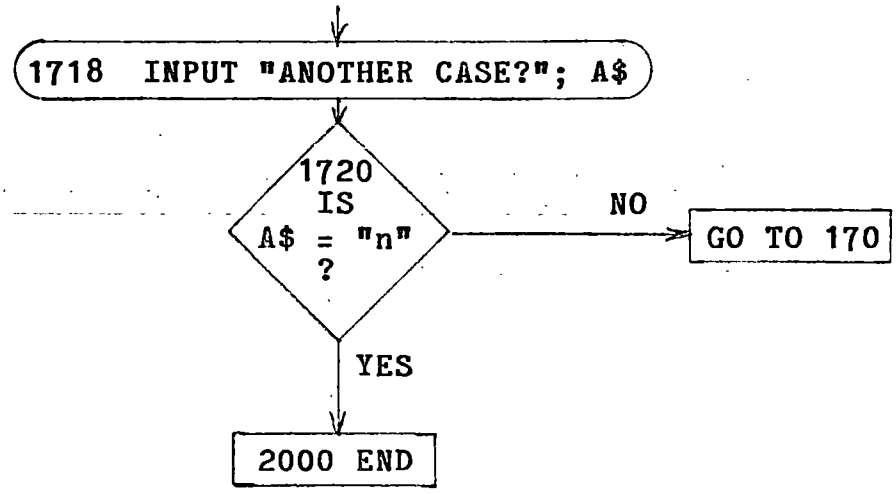

940 - 950 V = BK/WBAR * [1/COS(TH) - 1/COS(AO(I))] + LK * (BK^2)/(2 * WBAR) * [1/COS(TH)^2 - 1/COS(AO(I))^2]
960 YO = YC - V
970 IS |YO - YBAR| < .01 ?
YES
NO
980 IS NK > 1 ?
YES
NO
990 U1 = YO
1000 V1 = BT(I)
1010 NK = NK + 1
1020 DT = ATN((YO - YBAR / (XS(I) - XBAR))
1030 TH = BT(I) + DT
1040 BK = B(I) * COS(TH)
1050 GO TO 560
1070 U2 = U1
1080 V2 = V1
1090 U1 = YO
1100 V1 = BT(I)
1110 SL = (U2 - U1) / (V2 - V1)
1120 UB = U2 - SL * V2
1130 TH = (YBAR - UB) / SL
TO 1170

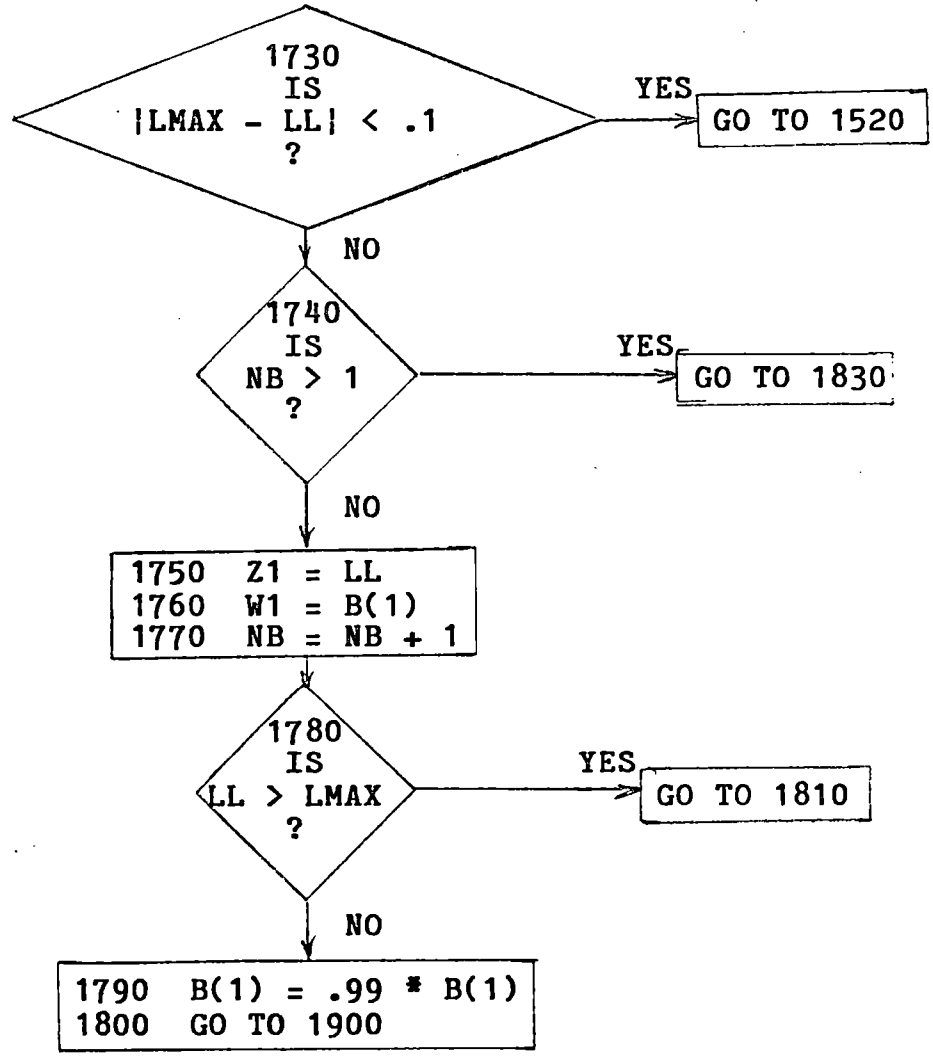

1170 A(I) = BK / COS(AO(I))
1180 IS J > 1 ?
YES
NO
1190 TL(I) =(B(I) * SIN(BT(I)) - A(I) * SIN(AO(I))) / WBAR
1191 IS I > 1 ?
YES
NO
1192 S(I) = LS - SS(I) + TL(I)
1193 GO TO 1210
1200 S(I) = TL(I) + S(I - 1)
1210 SS(I + 1) = SS(I) - TL(I)
1220 GO TO 1260
1230 S(I) = (TB(K - 1) * SIN(BA(K - 1)) - A(I) * SIN(AO(I))) / WBAR
1240 TL(I) = SS(I) + (J - 2) * LS + S(I)
1250 SS(I + 1) = LS - S(I)
1260 N(I) = J - 1
1270 LL = LL + TL(I)
1280 IS I = M ?
YES
NO
1290 I = I + 1
1300 B(I) = A(I - 1)
1310 KSTART = K
1320 GO TO 510
TO 1330

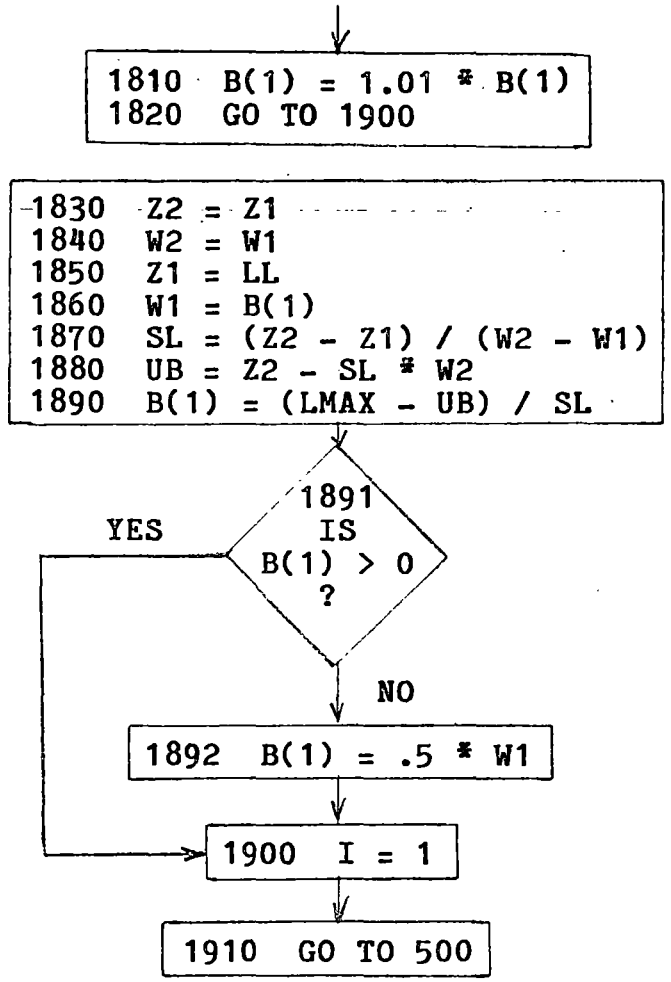

1330 IS PSS > 1 ?
YES
GO TO 1730
NO
1340 IS LL < LMAX ?
YES
GO TO 1370
NO
1350 LMAX = LL
1360 CS = SC
1370 INPUT "WANT INTERIM PRINTOUT?"; A$
1380 IS A$ = "n" ?
NO
GO TO 1520
YES
1390 INPUT "ANOTHER SET OF LOAD POSITIONS?"; A$
1400 IS A$ = "n" ?
NO
GO TO 450
YES
1410 - 1430 PRINT LMAX, CS
1440 - 1450 INPUT "ARE YOU SURE YOU DON'T WANT ANOTHER SET OF LOAD POSITIONS?"; A$
TO 1460


1460
IS
A$ = "y"
?
YES
GO TO 1490
NO
1470 PRINT "OK, ANOTHER SET OF LOAD POSITIONS"
1480 GO TO 450
1490 PSS = PSS + 1
1500 PRINT "BEGIN SECOND PASS"
1510 GO TO 450
1520 K = 1
1525 FOR I = 1 TO M STEP 1
1530 - 1550 PRINT I, SS(I), XS(I), YS(I), B(I), BT(I)
1560
IS
N(I) = 0
?
YES
NO
1570 FOR J = 1 TO N(I) STEP 1
1580 - 1610 PRINT J, WT(K), X(K), Y(K), TA(K),
AA(K), TB(K), BA(K)
1615 K = K + 1
1620 NEXT J
TO 1630


1707 GO TO 291
1708 INPUT "ARE YOU SURE?"; A$
1709 IS A$ = "y" ?
NO
GO TO 1705
YES
1710 INPUT "DIFFERENT CABLE LENGTH?"; A$
1711 IS A$ = "n" ?
YES
GO TO 1716
NO
1712 PRINT "CURRENT LENGTH ="; LMAX
1713 INPUT "WHAT IS NEW LENGTH?"; LMAX
1714 INPUT "SAME SET OF LOADS?"; A$
1715 IS A$ = "y" ?
YES
GO TO 450
NO
GO TO 1706
1716 INPUT "PASS 1, DIFFERENT TENSION?"; A$
1717 IS A$ = "y"
YES
GO TO 215
NO
TO 1718


1718 INPUT "ANOTHER CASE?"; A$
1720 IS A$ = "n" ?
NO
GO TO 170
YES
2000 END


1730 IS |LMAX - LL| < .1 ?
YES
GO TO 1520
NO
1740 IS NB > 1 ?
YES
GO TO 1830
NO
1750 Z1 = LL
1760 W1 = B(1)
1770 NB = NB + 1
1780 IS LL > LMAX ?
YES
GO TO 1810
NO
1790 B(1) = .99 * B(1)
1800 GO TO 1900


1810 B(1) = 1.01 * B(1)
1820 GO TO 1900
1830 Z2 = Z1
1840 W2 = W1
1850 Z1 = LL
1860 W1 = B(1)
1870 SL = (Z2 - Z1) / (W2 - W1)
1880 UB = Z2 - SL * W2
1890 B(1) = (LMAX - UB) / SL
1891 IS B(1) > 0 ?
YES
NO
1892 B(1) = .5 * W1
1900 I = 1
1910 GO TO 500

```
110 P!(1)=3.1415926536
120 INPUT"CABLE DIAMETER";DIA
140 INPUT"CABLE  MODULUS";E
150 LK!(1)=4/(P!(1)*DIA*DIA*E)
160 WBAR!(1)=1.035*((DIA/.75)**2)
170 INPUT"HOW MANY SPANS?";M
180 FORI=1TO(M+1)STEP1
190 PRINT"INPUT COORDINATES FOR I=";I
200 INPUT"XS(I),YS(I)=";XS(I),YS(I)
210 NEXTI
215 INPUT"ALLOWABLE CABLE TENSION=";ALL
220 INPUT"LOAD SPACING=";LS
221 INDEX=0
250 CH=0
260 FORI=1TOMSTEP1
270 CH=CH+SQR((XS(I+1)-XS(I))**2+(YS(I+1)-YS(I))**2)
280 NEXTI
290 MK=FIX(CH/LS+2)
291 INPUT"WANT ALL LOADS IDENTICAL?(y/n)";A$
292 IFA$="n"THEN350ELSE300
300 INPUT"WHAT LOAD?";WW
310 FORK=1TOMKSTEP1
320 W(K)=WW
330 NEXTK
340 GOTO430
350 PRINT"YOU'LL NEED ABOUT";MK;"LOADS"
355 INPUT"FOR K BETWEEN LOWER LIMIT OF";KL
360 INPUT"AND UPPER LIMIT OF";KU
370 INPUT"EACH LOAD IS";WW
380 FORK=KLTOKUSTEP1
390 W(K)=WW
400 NEXTK
410 INPUT"MORE LOADS TO ENTER?(y/n)";A$
420 IFA$="n"THEN430ELSE355
430 IFINDEX>0THEN450
431 PSS=1
440 LMAX!(1)=0
450 I=1
460 B!(I)=ALL
470 INPUT"CABLE LENGTH TO 1ST LOAD=";SS!(I)
480 SC!(1)=SS!(I)
490 NB=1
500 KSTART=1
505 LL!(1)=0
510 NK=1
520 TH!(1)=ATN((YS(I)-YS(I+1))/(XS(I)-XS(I+1)))
530 BK!(1)=B!(I)*COS(TH!(1))
540 XBAR=XS(I+1)
550 YBAR=YS(I+1)
560 J=1
570 K=KSTART
580 XC!(1)=XS(I)
590 YC!(1)=YS(I)
600 LBAR!(1)=SS!(I)
610 BT!(I)=TH!(1)
620 T!(1)=ATN(TAN(TH!(1))-WBAR!(1)*LBAR!(1)/BK!(1))
630 H!(1)=(1+SIN(TH!(1)))*COS(T!(1))/(COS(TH!(1))*(1+SIN(T!(1))))
640 H!(1)=LK!(1)*BK!(1)*LBAR!(1)+(BK!(1)/WBAR!(1))*LN(H!(1))
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650 V!(1)=1+.5*LK!(1)*BK!(1)*(1/COS(TH!(1))+1/COS(T!(1)))
660 V!(1)=BK!(1)*V!(1)*(1/COS(TH!(1))-1/COS(T!(1)))/WBAR!(1)
670 XO!(1)=XC!(1)-H!(1)
680 IFXO!(1)(XBARTHEN840
690 YO!(1)=YC!(1)-V!(1)
700 X!(K)=XO!(1)
710 Y!(K)=YO!(1)
720 TA!(K)=BK!(1)/COS(T!(1))
730 AA!(K)=T!(1)
740 WT(K)=W(K)
750 TH!(1)=ATN(TAN(T!(1))-WT(K)/BK!(1))
760 BA!(K)=TH!(1)
770 TB!(K)=BK!(1)/COS(TH!(1))
780 XC!(1)=X!(K)
790 YC!(1)=Y!(K)
800 LBAR!(1)=LS
810 J=J+1
820 K=K+1
830 GOTO620
840 H!(1)=XC!(1)-XBAR
850 T!(1)=TH!(1)
860 F1!(1)=(COS(TH!(1))/(1+SIN(TH!(1))))**2
870 F2!(1)=2*(WBAR!(1)*H!(1)/BK!(1)-LK!(1)*BK!(1)*TAN(TH!(1)))
880 F3!(1)=2*LK!(1)*BK!(1)*TAN(T!(1))
890 F!(1)=F1!(1)*EXP(F2!(1)+F3!(1))
900 AO!(I)=ATN((1-F!(1))/(2*SQR(F!(1))))
910 IFABS(AO!(I)-T!(1))(.0000001THEN940
920 T!(1)=AO!(I)
930 GOTO880
940 V!(1)=1+.5*LK!(1)*BK!(1)*(1/COS(TH!(1))+1/COS(AO!(I)))
950 V!(1)=BK!(1)*V!(1)*(1/COS(TH!(1))-1/COS(AO!(I)))/WBAR!(1)
960 YO!(1)=YC!(1)-V!(1)
970 IFABS(YO!(1)-YBAR)(.01THEN1170
980 IFNK)1THEN1070
990 U1!(1)=YO!(1)
1000 V1!(1)=BT!(I)
1010 NK=NK+1
1020 DT!(1)=ATN((YO!(1)-YBAR)/(XS(I)-XBAR))
1030 TH!(1)=BT!(I)+DT!(1)
1040 BK!(1)=B!(I)*COS(TH!(1))
1050 GOTO560
1070 U2!(1)=U1!(1)
1080 V2!(1)=V1!(1)
1090 U1!(1)=YO!(1)
1100 V1!(1)=BT!(I)
1110 SL!(1)=(U2!(1)-U1!(1))/(V2!(1)-V1!(1))
1120 UB!(1)=U2!(1)-SL!(1)*V2!(1)
1130 TH!(1)=(YBAR-UB!(1))/SL!(1)
1140 GOTO1040
1170 A!(I)=BK!(1)/COS(AO!(I))
1180 IFJ)1THEN1230
1190 TL!(I)=(B!(I)*SIN(BT!(I))-A!(I)*SIN(AO!(I)))/WBAR!(1)
1191 IFI)1THEN1200
1192 S!(I)=LS-SS!(I)+TL!(I)
1193 GOTO1210
1200 S!(I)=TL!(I)+S!(I-1)
1210 SS!(I+1)=SS!(I)-TL!(I)
1220 GOTO1260
1230 S!(I)=(TB!(K-1)*SIN(BA!(K-1))-A!(I)*SIN(AO!(I)))/WBAR!(1)
1240 TL!(I)=SS!(I)+(J-2)*LS-S!(I)
```

```
1250 SS!(I+1)=LS-S!(I)
1260 N(I)=J-1
1270 LL!(1)=LL!(1)+TL!(I)
1280 IFI=MTHEN1330
1290 I=I+1
1300 B!(I)=A!(I-1)
1310 KSTART=K
1320 GOTO510
1330 IFPSS>1THEN1730
1340 IFLL!(1)<LMAX!(1)THEN1370
1350 LMAX!(1)=LL!(1)
1360 CS!(1)=SC!(1)
1370 INPUT"WANT INTERIM PRINTOUT?(y/n)";A$
1380 IFA$="n"THEN1390ELSE1520
1390 INPUT"ANOTHER SET OF LOAD POSITIONS?(y/n)";A$
1400 IFA$="n"THEN1410ELSE450
1410 PRINT"MAXIMUM TOTAL CABLE LENGTH=";LMAX!(1)
1420 PRINT"OCCURS WHEN CABLE LENGTH FROM UPPERMOST"
1430 PRINT"SUPPORT TO FIRST LOAD IS";CS!(1);"FEET"
1440 PRINT"ARE YOU SURE YOU DON'T WANT"
1450 INPUT"ANOTHER SET OF LOAD POSITIONS?(y/n)";A$
1460 IFA$="y"THEN1490
1470 PRINT"O.K., TRY ANOTHER SET OF POSITIONS"
1480 GOTO450
1490 PSS=PSS+1
1500 PRINT"BEGIN SECOND PASS"
1510 GOTO450
1520 K=1
1525 FORI=1TOMSTEP1
1530 PRINT"I,SS(I)=";I;SS!(I)
1540 PRINT"XS(I),YS(I)=";XS(I);YS(I)
1550 PRINT"B(I),BT(I)=";B!(I);BT!(I)
1555 INPUT"HIT ENTER TO CONTINUE";A$
1560 IFN(I)=0THEN1630
1570 FORJ=1TON(I)STEP1
1580 PRINT"J,WT(I,J)=";J;WT(K)
1590 PRINT"X(I,J),Y(I,J)=";X!(K);Y!(K)
1600 PRINT"TA(I,J),AA(I,J)=";TA!(K);AA!(K)
1610 PRINT"TB(I,J),BA(I,J)=";TB!(K);BA!(K)
1615 K=K+1
1616 INPUT"HIT ENTER TO CONTINUE";A$
1620 NEXTJ
1630 PRINT"TL(I),S(I)=";TL!(I);S!(I)
1640 PRINT"A(I),AO(I)=";A!(I);AO!(I)
1645 INPUT"HIT ENTER TO CONTINUE";A$
1650 NEXTI
1660 PRINT"XS(M+1),YS(M+1)=";XS(M+1);YO!(1)
1670 PRINT"TOTAL UNSTRETCHED LENGTH=";LL!(1)
1675 PRINT"SS(M+1)=";SS!(M+1)
1680 IFPSS=1THEN1390
1690 INPUT"ANOTHER SET OF LOAD POSITIONS?(y/n)";A$
1700 IFA$="n"THEN1701ELSE450
1701 INPUT"ARE YOU SURE?(y/n)";A$
1702 IFA$="y"THEN1703ELSE1470
1703 INPUT"WANT DIFFERENT SET OF LOADS?(y/n)";A$
1704 IFA$="n"THEN1708ELSE1706
1705 PRINT"O.K.,NEW SET OF LOADS"
1706 INDEX=1
1707 GOTO291
1708 INPUT"ARE YOU SURE?(y/n)";A$
```

```
1709 IFA$="y"THEN1710ELSE1705
1710 INPUT"WANT DIFFERENT CABLE LENGTH?(y/n)";A$
1711 IFA$="n"THEN1716
1712 PRINT"CURRENT LENGTH=";LMAX!(1)
1713 INPUT"WHAT IS NEW LENGTH?";LMAX!(1)
1714 INPUT"WANT SAME SET OF LOADS?(y/n)";A$
1715 IFA$="y"THEN450ELSE1706
1716 INPUT"PASS 1,DIFFERENT TENSION?(y/n)";A$
1717 IFA$="y"THEN215
1718 INPUT"WANT ANOTHER CASE?(y/n)";A$
1720 IFA$="n"THEN2000ELSE170
1730 IFABS(LMAX!(1)-LL!(1))<.1THEN1520
1740 IFNB>1THEN1830
1750 Z1!(1)=LL!(1)
1760 W1!(1)=B!(1)
1770 NB=NB+1
1780 IFLL!(1)>LMAX!(1)THEN1810
1790 B!(1)=.99*B!(1)
1800 GOTO1900
1810 B!(1)=1.01*B!(1)
1820 GOTO1900
1830 Z2!(1)=Z1!(1)
1840 W2!(1)=W1!(1)
1850 Z1!(1)=LL!(1)
1860 W1!(1)=B!(1)
1870 SL!(1)=(Z2!(1)-Z1!(1))/(W2!(1)-W1!(1))
1880 UB!(1)=Z2!(1)-SL!(1)*W2!(1)
1890 B!(1)=(LMAX!(1)-UB!(1))/SL!(1)
1891 IFB!(1)>0THEN1900
1892 B!(1)=.5*W1!(1)
1900 I=1
1910 GOTO500
2000 END
```

ATTACHMENT 4

DRAFT MANUSCRIPT

"IMPROVED THEORY FOR STATIC ANALYSIS OF CABLE SYSTEMS"

IMPROVED THEORY FOR STATIC ANALYSIS OF CABLE SYSTEMS

by

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BACKGROUND AND PURPOSE

In some types of cable systems, the unstretched length of cable in a span is fixed. So-called standing skyline systems are of this type. The skyline in such a system is anchored at both extremities, and it may or may not be supported at intermediate locations. Even though its unstretched length remains unchanged, its stretched length will vary as load magnitudes and locations change.

We have been investigating the potential for a long-distance cable transportation system, patterned after the Japanese Zig-Zag System (Moroto and Fujiwara, 1979) in which cable tensions are controlled at intermediate locations along its route by specially designed mechanisms. Design and analysis of such a system depends upon the ability to account for unstretched as well as stretched cable lengths in the segments between control mechanisms.

Carson (1970) and Carson and Emery (1975) necessarily dealt with stress-strain relationships in studying the dynamic behavior of cables. Nevertheless they relied on conventional catenary theory to determine initial strain energy and to approximate cable shape during vibration.

Carson (1977) later provided a rigorous analytic treatment of the static characteristics of a single cable segment, in which he showed how to solve several types of problems. Included were problems entailing cable length; still, he used conventional catenary theory, making no distinction between stretched and unstretched length.

Conventional catenary theory does not account for cable strain and is based on the assumption that the cable's weight per unit length is invariant. This assumption will not introduce serious errors in estimates of cable system tensions or load capacities if cable length is not a consideration. However, if a cable length criterion is to be satisfied in analysis, conventional catenary theory is inappropriate.

The purpose of this paper is to describe a more suitable theory for cable analysis and to compare it with conventional catenary theory. In addition, a modified catenary theory is described that may be advantageous in some situations.

THEORETICAL DEVELOPMENT

Figure 1 illustrates a cable element of differential length (ds). It also shows the tensions acting on the element (T and T + dT) and the angles between these tension vectors and horizontal (θ and $\theta + d\theta$).

If the weight per unit length of the cable element is w, then the equation for vertical equilibrium can be written

1. $(T + dT) \sin (\theta + d\theta) - T \sin \theta - wds = 0$

and, for horizontal equilibrium,

2. $(T + dT) \cos (\theta + d\theta) - T \cos \theta = 0$

Letting ε represent the cable strain, we can write

3. $$w = \frac{\bar{w}}{1 + \varepsilon}$$

where $\bar{w}$ is the weight per unit length of the cable in a relaxed or unstretched state.

If ϕ is the nominal diameter of the cable and E is its tensile modulus of elasticity, then

4. $$\varepsilon = \frac{4T}{\pi\phi^2 E}$$

Accordingly, by combining equations 3 and 4, we can write

5. $$w = \frac{\bar{w}}{1 + \frac{4T}{\pi\phi^2 E}}$$

If we let

$$6. \quad k = \frac{4}{\pi \phi^2 E}$$

then equation 5 becomes

$$7. \quad w = \frac{\bar{w}}{1 + kT}$$

By combining equations 1 and 7, substituting appropriate trigonometric identities, rearranging terms, letting cos $d\theta$ = 1 and sin $d\theta$ = $d\theta$, and ignoring terms containing products of two or more differential quantities, it can be shown that

$$8. \quad T \cos \theta \, d\theta + \sin \theta \, dT - \frac{\bar{w}}{1 + kT} ds = 0$$

Likewise, equation 2 becomes

$$9. \quad T \sin \theta \, d\theta - \cos \theta \, dT = 0$$

By rearranging and integrating equation 9, it can be shown that

$$10. \quad T \cos \theta = H$$

where H is a constant. (Note that H is the horizontal component of cable tension at any location in a span.)

Therefore, by combining equations 8 and 10, it can be shown that

$$11. \quad H \, d\theta + H \tan^2 \theta \, d\theta - \frac{\bar{w}}{1 + \frac{kH}{\cos \theta}} ds = 0$$

By rearranging equation 11 and making appropriate trigonometric substitutions, we have

$$12. \quad ds = \frac{H}{\bar{w}} \left[\frac{1}{\cos^2 \theta} + \frac{kH}{\cos^3 \theta}\right] d\theta$$

Equation 12 can be integrated to yield

$$13.\quad (s - s_0) = \frac{H}{\bar{w}} \{\tan\theta - \tan\theta_0 + \frac{k H}{2} [\ln (\frac{1 + \sin\theta}{\cos\theta}) + \frac{\tan\theta}{\cos\theta} - \ln (\frac{1 + \sin\theta_0}{\cos\theta_0}) - \frac{\tan\theta_0}{\cos\theta_0}]\}$$

where $(s - s_0)$ is the actual (or stretched) length of an uninterrupted cable segment, one end of which is under a tension T_0 directed at an angle θ_0 with the horizontal, and the other end of which is under a tension T directed at an angle θ with the horizontal; and where

$$14.\quad T \cos\theta = T_0 \cos\theta_0 = H$$

Figure 2 illustrates such a cable segment.

From Figure 2, we can write the equation of vertical equilibrium as

$$15.\quad T \sin\theta - T_0 \sin\theta_0 - W_c = 0$$

where W_c is the weight of the cable suspended between its two extremities. Note that, if the weight per unit length of unstretched cable is $\bar{w}$, as previously defined, then it must be true that the unstretched length of cable ($\bar{L}$) between the suspension points in Figure 2 is

$$16.\quad \bar{L} = W_c / \bar{w}$$

Accordingly, combining equations 15 and 16, we have

$$17.\quad \bar{L} = \frac{T \sin\theta - T_0 \sin\theta_0}{\bar{w}}$$

By combining equations 13, 14 and 17, it can be shown that

$$18.\quad (s - s_0) - \bar{L} = \frac{kH^2}{2\bar{w}} \left[\ln \left(\frac{1 + \sin \theta}{\cos \theta}\right) + \frac{\tan \theta}{\cos \theta} - \ln \left(\frac{1 + \sin \theta_0}{\cos \theta_0}\right) - \frac{\tan \theta_0}{\cos \theta_0}\right]$$

Equation 18 expresses the difference between the stretched length $(s - s_0)$ and unstretched length $(\bar{L})$ of cable in a span.

It should be noted that conventional catenary theory does not enable distinction between stretched and unstretched cable length. Rather, a single length prediction is provided, the equation for which is identical in form to that for $\bar{L}$ in the above development (equation 17). However, even though the form of the length prediction equation derived from conventional catenary theory is identical to equation 17, the predicted length will not equal $\bar{L}$ because, ceteris paribus, the tensions (T) and slopes (θ) predicted by the two theories are different.

The differences between these theories will become clearer through examples presented later. Some further manipulations are needed, however, to facilitate certain computations.

Returning to equation 13 and making use of the relationships $dx = ds \cos \theta$ and $dy = ds \sin \theta$, we can determine by differentiation that

$$19.\quad \frac{dx}{d\theta} = \frac{H}{\bar{w}} \left[\frac{1}{\cos \theta} + \frac{kH}{\cos^2 \theta}\right]$$

and

$$20.\quad \frac{dy}{d\theta} = \frac{H}{\bar{w}} \tan \theta \left[\frac{1}{\cos \theta} + \frac{kH}{\cos^2 \theta}\right]$$

By rearranging and integrating between appropriate limits, equations 19 and 20 can be shown to yield, respectively,

$$21.\quad (x - x_0) = \frac{H}{\bar{w}} \{kH\ (\tan\theta - \tan\theta_0) + \ln\ [(\frac{1 + \sin\theta}{\cos\theta})(\frac{\cos\theta_0}{1 + \sin\theta_0})]\}$$

and

$$22.\quad (y - y_0) = \frac{H}{\bar{w}} [\frac{1}{\cos\theta} - \frac{1}{\cos\theta_0}][1 + \frac{kH}{2}(\frac{1}{\cos\theta} + \frac{1}{\cos\theta_0})]$$

where $(x - x_0)$ and $(y - y_0)$ are the differences between the horizontal and vertical coordinates, respectively, of the extremities of an uninterrupted cable segment such as that shown in Figure 2.

Equation 21 can be rearranged and transformed into

$$23.\quad \theta_0 = \sin^{-1}(\frac{1 - f}{1 + f})$$

where

$$23a.\quad f = \frac{\cos^2\theta}{(1 + \sin\theta)^2} e^{\psi}$$

and

$$24.\quad \psi = 2\ [\frac{\bar{w}}{H}(x - x_0) - kH\ (\tan\theta - \tan\theta_0)]$$

Note that f is a function of θ_0, so equation 23 cannot be solved directly for θ_0. Rather, an iterative routine must be employed; more about which will be discussed later.

Alternatively, equation 21 can be rearranged and transformed into

$$25.\quad \theta = \sin^{-1}(\frac{g - 1}{g + 1})$$

where

$$25a.\quad g = \frac{(1 + \sin\theta_0)^2}{\cos^2\theta_0} e^{\psi}$$

Thus, depending on whether θ or θ_0 is given (along with H and $x - x_0$), equation 23 or equation 25 can be used iteratively to find θ_0 or θ, respectively.

COMPARISON WITH CONVENTIONAL CATENARY THEORY

Perhaps the simplest comparison with conventional catenary theory can be made by analyzing a cable extending indefinitely outward from an anchor point at which the slope of the cable, θ_0, is zero. Then the values of H and T_0 will be identical. Such a cable is illustrated in Figure 3.

Conventional catenary theory yields the following relationships with respect to Figure 3 (e.g. see Reddick and Miller, 1955):

$$26. \quad \theta = \cos^{-1}\left[\frac{1}{\cosh\left(\frac{w\,x}{H}\right)}\right] = \sin^{-1}\left[\tanh\left(\frac{w\,x}{H}\right)\right]$$

$$27. \quad T = H \cosh\left(\frac{w\,x}{H}\right) = H/\cos\theta$$

$$28. \quad y = \frac{H}{w}\left[\cosh\left(\frac{w\,x}{H}\right) - 1\right] = \frac{H}{w}\left[\frac{1}{\cos\theta} - 1\right]$$

$$29. \quad s = \frac{H}{w}\sinh\left(\frac{w\,x}{H}\right) = \frac{H}{w}\tan\theta = \frac{T\sin\theta}{w}$$

where s is the catenary length from the anchor point to the coordinates (x, y), w is the weight per unit length of the cable, and the remaining variables are as shown in Figure 3. Suppose, for example, the cable is nominally 0.75 inch in diameter with an unstretched weight per unit length, $\bar{w}$ = 1.035 p.p.f. If we assume T_0 = H = 20,000 lbs. and w = $\bar{w}$, then the values for θ, T, y and s (obtained from equations 26 - 29, respectively) at various locations, x, will be as shown in Table I.

To obtain comparable results with the theory derived in the previous section (hereafter called the "exact" theory), it is necessary to use an iterative scheme to find θ for given values of x, employing equations 24, 25 and 25a.

For example, suppose (as above) that the cable diameter, ϕ, is 0.75 inch and its modulus, E, is 10×10^6 p.s.i*. Then, from equation 6, $k = 4/\pi\phi^2 E = 2.263537 \times 10^{-7}$ lb.$^{-1}$. With $\bar{w} = 1.035$ p.p.f., $\theta_0 = 0$, and H = 20,000 lbs., suppose we seek the value of θ at x = 1000 ft. If we start with an assumption that $\theta = \theta_0 = 0$, then equation 24 will yield $\psi = 0.1035$; so equation 25a will yield g = 1.109046 and, from equation 25, we find $\theta = 2.963734°$. Since this value of θ is obviously different from the value for θ that was assumed to initiate the solution (recall we started by assuming $\theta = 0$), we repeat the solution starting with the value of θ just calculated, which is 2.963734°.

Therefore, starting with $\theta = 2.963734°$, we find from equation 24 that $\psi = 0.103031$, and from equation 25a that g = 1.108526; so from equation 25 we find $\theta = 2.950323°$.

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*Published values of E for many types of wire rope range between 10×10^6 and 20×10^6 p.s.i. However, wire rope strain is based on stress computations that employ the metallic cross-sectional area, not the nominal cable cross-sectional area as used herein. Because the metallic area is in the vicinity of one-half the nominal cable area, the value for $E = 10 \times 10^6$ p.s.i. used herein is probably conservative. That is, a value between 5×10^6 and 10×10^6 might be more accurate. (Reference: Wire Rope Engineering Handbook. American Steel and Wire Company of New Jersey. 1943. 136 pp., illus.)

Iterating again, this time with a starting value of $\theta = 2.950323°$, we find $\psi = 0.103033$, $g = 1.108528$, and $\theta = 2.950384°$. Another iteration, starting with $\theta = 2.950384°$, results in values of $\psi = 0.103033^*$, $g = 1.108528^*$ and $\theta = 2.950383°$; and yet another iteration results in negligible difference between the assumed and calculated values of θ (at least to the sixth place beyond the decimal point).

In general, fewer than five iterations are needed to obtain values of θ that are within .0000001 degree of the "correct" value. (Note that, if θ is given and it is desired to find θ_0, then equations 23a and 24 would be used in a similar iterative scheme).

For the cable shown in Figure 3, using values of $\bar{w} = 1.035$ p.p.f., $H = 20{,}000$ lbs., and $E = 10 \times 10^6$ p.s.i., we find from equations 25 (iteratively with equations 25a and 24), 10 (rearranged), 22, 13, and 17, respectively, the values for θ, T, y, s, and $\bar{L}$ shown in Table II. (Note that, in applying equations 13, 17, 22, 24, and 25a to Figure 3, the lower limits of θ_0, x_0, y_0 and s_0 are all zero.)

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*These values of ψ and g appear not to change from the previous iteration. In fact they change slightly, but rounding to the sixth place beyond the decimal point masks their change.

A comparison of Tables I and II shows that, except for extremely long spans, the differences in values of θ, T, y and s predicted by the two theories are trivial. Indeed, considering the need for an iteration scheme to find the values of θ with the exact theory, one might conclude that the conventional catenary theory is preferable in the given application. Note, however, that the conventional catenary theory provides no estimate of unstretched cable length; whereas the exact theory predicts the stretched cable length, s, as well as its unstretched length, $\bar{L}$. To better appreciate the importance of this difference between the two theories, consider the following example.

Assume a 3/4-inch diameter cable is suspended between two supports that are separated 1000 feet horizontally and 400 feet vertically; and suppose that its length is adjusted so that, when a 4000 lb. load is attached at mid-span, its tension at the uppermost support is 20,000 lbs. Of interest are the differences between the two theories being considered, given the criterion of constant unstretched cable length between the supports, as the load magnitude and location is changed. Figure 4 illustrates the situation and defines the load coordinates (x_c and y_c) and support tensions (A and B).

In the following, when using the exact theory, a value for E of 10×10^6 p.s.i. is assumed, resulting in a value for k (from equation 6) of 2.263537×10^{-7} lb.$^{-1}$.

From conventional catenary theory, a cable length of 1,083.8157 ft. results when a 4,000 lb. load is attached at mid-span and B = 20,000 lbs. Therefore, because no distinction between stretched and unstretched length can be made with conventional catenary theory, all the results in Tables III, V, and VII satisfy the criterion that cable length = 1,083.8157 ± .001* ft. Similarly, from the exact theory, a stretched cable length of 1083.8074 ft.--but an unstretched length of 1,079.1521 ft.--results when a 4,000 lb. load is attached at mid-span and B = 20,000 lbs. Consequently, all the results in Tables IV, VI, and VIII satisfy the criterion that unstretched cable length = 1.079.1521 ± .001* ft.

A comparison of Tables III and IV shows that, as the 4000 lb. load is moved (i.e. as x_c changes), the differences between support tensions (A and B) and load elevation (y_c) predicted by the two theories become greater as the load nears either support. When the load is at midspan there is virtually no difference in predictions (which should be expected, because the criterion that B = 20,000 lbs. when the 4,000 lb. load is at mid-span was satisfied with each theory). As the load is moved toward either support, the stretched cable length predicted by the exact theory diminishes, and the resulting load elevation (y_c) is predicted to be higher than that predicted by the conventional catenary theory. Correspondingly, the support tensions (A and B) are predicted by the exact theory to be higher than those predicted by the

---

*Regardless of the theory used, iterative computation methods are necessary to satisfy length criteria; and in our computer routines we arbitrarily chose tolerance limits of ±.001 ft.

conventional catenary theory as the load approaches the supports. Of course, the conventional catenary theory provides no separate predictions of stretched and unstretched lengths, as previously noted.

If the load is halved, from 4000 to 2000 lbs., more striking differences between the theories become evident, as shown by Tables V and VI. At all values of x_c are the load elevations (y_c) and support tensions (A and B) predicted to be higher by the exact theory than by the conventional catenary theory. More important is the credibility of predictions. A comparison of Tables III and V shows only a slight difference in load elevation (y_c) when the loads are at midspan (i.e. y_c = 135.031 ft. when W = 4000 lbs. versus y_c = 135.378 ft. when W = 2000 lbs.). In contrast, a comparison of Tables IV and VI shows a considerably greater (and more plausible) difference in y_c when the loads are at midspan (i.e. y_c = 135.070 ft. when W = 4000 lbs. versus y_c = 143.798 ft. when W = 2000 lbs.).

Consider next the consequence of doubling the load from 4000 lbs. to 8000 lbs. Tables VII and VIII contrast the two theories, showing that the load elevations (y_c) and tensions (A and B) are generally predicted to be lower by the exact theory than by the conventional catenary theory. Again, plausibility can be assessed by comparing Table III with Table VII and by comparing Table IV with Table VIII, whereby it will be noted that doubling the load results in only slight change in load elevation (y_c) predicted by conventional catenary theory when the loads are at midspan (i.e. y_c = 135.031 ft. when W = 4000 lbs. versus y_c = 134.916 ft. when W = 8000 lbs.); whereas with

the exact theory the change in predicted midspan deflection is substantial (i.e. y_c = 135.070 ft. when W = 4000 lbs. versus y_c = 122.928 ft. when W = 8000 lbs.).

MODIFIED CATENARY THEORY

It is apparent from the foregoing that conventional catenary theory is deficient for analyses wherein stretched versus unstretched cable length is a primary consideration. However, it is also apparent that, for some analyses, solutions employing the conventional catenary theory require fewer iterative computation procedures. Therefore, it would be desirable to have a theory and computation procedures that reflect the advantages of both theories--the accuracy of the exact theory and the computational facility of the conventional catenary theory.

Regardless of which theory is employed or how many loads are applied in a span, computational schemes usually entail consideration of individual, unloaded cable segments. Often the starting point in any scheme is to assume a tension and slope at one support, then compute the tension, slope, and elevation of the cable at the next location where a load is applied (which will be the next support if no load is applied in a span). Equilibrium considerations provide the basis for determining the tension and slope in the next cable segment at the load juncture, and solution then proceeds to the next load application point (or support, if no additional loads are encountered). Ultimately, the predicted cable coordinates must match the actual coordinates of the next support to a close approximation. If not, the scheme must be repeated, starting with a different assumption regarding tension, slope, or both.

If, based on the tension assumed (or calculated) at one end of an unloaded cable segment (T) the cable weight per unit length is modified, namely

$$30. \quad w = \frac{\bar{w}}{1 + \frac{4T}{\pi\phi^2 E}}$$

then it should be possible to rely on conventional catenary theory. That is, for any unloaded cable segment, equation 30 can be combined with equations 26 through 29 to obtain estimates of the tensions, slopes, coordinates and stretched cable length; then equation 17 can be used to obtain the unstretched cable length.

Tables IX, X, and XI result from applying the modified catenary theory to the situation represented in Figure 4. A comparison of Tables IX, X and XI with Tables IV, VI, and VIII, respectively, shows reasonably close agreement between the exact and modified theories. (Note that the unstretched cable length resulting from the modified catenary theory--when a 4,000 lb. load is applied at mid-span and B = 20,000 lbs.--is 1,079.1263 ft. This is slightly different from the value of 1,079.1521 ft. that results from the exact theory because the latter accounts for the slight variation in weight per unit length of the cable along its length; whereas the modified catenary theory does not.)

Finally, if the modified catenary theory is applied to the situation portrayed in Figure 3, using the relationships

$$31. \quad w = \frac{\bar{w}}{1 + \frac{4T_0}{\pi\phi^2 E}}$$

and (for the unstretched cable length)

$$32. \quad \bar{L} = \frac{T \sin \theta}{\bar{w}}$$

together with equations 26 - 29, the results shown in Table XII are found.

A comparison of Tables II and XII shows relatively close agreement between the exact and modified catenary theories; and, like the exact theory, the modified catenary theory provides estimates of both the stretched and unstretched lengths, whereas conventional catenary theory does not.

DISCUSSION

It has been noted in the foregoing that, when the horizontal coordinate is specified (e.g. the value of x in Figure 3 or the value of x_c in Figure 4) then the exact theory is somewhat disadvantageous in that it requires certain iterative computations that the conventional and modified catenary theories do not. However, in certain types of problems, the unstretched length of cable from a support to a load (or from one load to the next load) is specified; in such cases it is necessary to compute the x and y coordinates rather than assuming or specifying them. The exact theory is well suited for such problems, as shown in the following.

By combining equations 14 and 17, it can be shown that

$$33. \quad \tan \theta_0 = \tan \theta - \frac{\bar{w}\,\bar{L}}{H}$$

Therefore, if θ, H and $\bar{L}$ are specified or assumed, θ_0 easily can be found from equation 33; after which equations 21 and 22 can be used to compute directly (without iteration) the horizontal and vertical coordinate changes from one end to the other end of the cable segment whose unstretched length is $\bar{L}$.

SUMMARY

A theory for describing the behavior of cables under static load conditions, incorporating stress-strain relationships and enabling prediction of both stretched and unstretched cable length, has been derived and compared with conventional catenary theory. Predictions based on this theory seem more plausible than those based on conventional catenary theory; and the differences in predictions are of sufficient magnitude to be readily testable. It is hoped that interested readers will conduct such tests.

In addition to the new theory derived herein, a means for modifying conventional catenary theory to provide comparable predictions has been suggested. For some types of problems this modified catenary theory may be deemed preferable to the new theory, in that certain computations can be made more directly (i.e. without the need for iterative numerical computations). However, for problems entailing specification of unstretched cable length--from which cable slopes, tensions, and coordinates are to be found--the new theory developed herein seems most suitable.

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Table I.--Results from Conventional Catenary Theory for Figure 3, $T_0 = H = 20{,}000$ lbs., $w = \bar{w} = 1.035$ p.p.f.

x (ft)	θ (deg)	T (lbs)	y (ft)	s (ft)
0	0	20,000.000	0	0
100	.29650	20,000.268	.2588	100.0004
250	.74124	20,001.674	1.6172	250.0070
500	1.48236	20,006.696	6.4691	500.0558
1,000	2.96373	20,026.787	25.8808	1,000.4464
2,500	7.39205	20,167.612	161.9444	2,506.9800
5,000	14.66257	20,673.259	650.4922	5,055.9800
10,000	28.40938	22,738.365	2,645.7636	10,452.3587

Table II.--Results from Exact Theory for Figure 3, ϕ = .75 in., $T_0 = H$ = 20,000 lbs., $\bar{w}$ = 1.035 p.p.f. and E = 10 x 10^6 p.s.i.

x (ft)	θ (deg)	T (lbs)	y (ft)	s (ft)	$\bar{L}$ (ft)
0	0	20,000.000	0	0	0
100	.29517	20,000.265	.2576	100.0004	99.5498
250	.73790	20,001.659	1.6099	250.0069	248.8802
500	1.47568	20,006.635	6.4400	500.0553	497.8015
1,000	2.95038	20,026.546	25.7641	1,000.4424	995.9317
2,500	7.35883	20,166.099	161.2116	2,506.9170	2,495.5880
5,000	14.59721	20,667.104	647.5117	5,055.4716	5,032.4344
10,000	28.28650	22,712.077	2,633.0316	10,448.1265	10,398.8676

Table III. Results from Conventional Catenary Theory for Figure 4, $w = \bar{w} = 1.035$ p.p.f., $W = 4000$ lbs., maximum $B = 20{,}000$ lbs., cable length = 1,083.8157 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)
50	-5.689	8,483.15	9,386.68
100	3.374	11,205.02	12,454.93
200	29.784	14,671.35	16,203.95
300	61.621	16,713.37	18,397.21
400	96.923	17,803.98	19,598.47
500	135.031	18,109.73	20,000.00
600	175.786	17,674.91	19,659.20
700	219.361	16,444.87	18,533.58
800	266.444	14,233.93	16,458.42
900	319.134	10,508.98	12,966.80
950	350.013	7,542.20	10,250.81

Table IV. Results from Exact Theory for Figure 4, ϕ = .75 in., $E = 10 \times 10^6$ p.s.i., $\bar{w}$ = 1.035 p.p.f., W = 4000 lbs., maximum B = 20,000 lbs., unstretched cable length = 1,079.1521 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)	Stretched cable length (ft)
50	-1.646	10,002.05	11,072.14	1081.794
100	7.138	12,505.73	13,826.12	1082.453
200	32.118	15,405.46	16,954.83	1083.185
300	62.646	17,014.97	18,701.23	1083.575
400	97.145	17,860.00	19,653.17	1083.767
500	135.070	18,111.51	20,000.00	1083.807
600	176.298	17,812.41	19,794.32	1083.708
700	221.012	16,910.31	18,992.63	1083.458
800	269.842	15,216.81	17,421.21	1083.010
900	324.570	12,157.26	14,550.14	1082.223
950	356.022	9,480.12	12,057.50	1081.546

Table V. Results from Conventional Catenary Theory for Figure 4, $w = \bar{w} = 1.035$ p.p.f., W = 2000 lbs., cable length = 1,083.8157 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)
50	-3.900	5.093.73	5,749.55
100	4.659	6,514.00	7,331.26
200	30.553	8,357.20	9,318.19
300	62.125	9,450.68	10,491.71
400	97.296	10,034.32	11,134.77
500	135.378	10,194.83	11,346.96
600	176.199	9,954.50	11,157.34
700	219.970	9,287.82	10,546.60
800	267.465	8,103.36	9,433.57
900	321.098	6,153.20	7,600.19
950	353.064	4,674.69	6,234.14

Table VI. Results from Exact Theory for Figure 4, ϕ = .75 in., $E = 10 \times 10^6$ p.s.i., $\bar{w}$ = 1.035 p.p.f., W = 2000 lbs., unstretched cable length = 1,079.1521 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)	Stretched cable length (ft)
50	1.784	6,661.82	7,444.14	1080.911
100	11.766	8,193.67	9,082.97	1081.305
200	38.553	10,029.13	11,022.72	1081.762
300	70.310	11,062.88	12,120.17	1082.011
400	105.559	11,607.54	12,714.88	1082.136
500	143.798	11,768.73	12,920.83	1082.164
600	184.911	11,573.24	12,769.32	1082.103
700	229.067	10,991.16	12,234.59	1081.946
800	276.853	9,912.80	11,213.74	1081.665
900	329.965	8,020.05	9,408.19	1081.184
950	360.349	6,460.57	7,928.43	1080.790

Table VII. Results from Conventional Catenary Theory for Figure 4, $w = \bar{w} = 1.035$ p.p.f., W = 8000 lbs., cable length = 1,083.8157 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)
50	-6.368	15,545.50	16,987.32
100	2.919	20,781.67	22,917.26
200	29.522	27,411.55	30,095.82
300	61.448	31,307.58	34,281.25
400	96.790	33,391.58	36,576.00
500	134.916	33,990.47	37,357.00
600	175.644	33,176.46	36,721.80
700	219.150	30,848.36	34,592.78
800	266.093	26,645.55	30,650.31
900	318.409	19,494.68	23,955.10
950	348.721	13,681.25	18,653.45

Table VIII. Results from Exact Theory for Figure 4, ϕ = .75 in., E = 10 x 10^6 p.s.i., $\bar{w}$ = 1.035 p.p.f., W = 8000 lbs., unstretched cable length = 1,079.1521 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)	Stretched cable length (ft)
50	-5.513	16,029.40	17,537.35	1083.369
100	1.326	19,914.42	21,990.43	1084.431
200	23.481	24,379.91	26,971.66	1085.585
300	52.128	26,848.55	29,748.94	1086.191
400	85.480	28,141.09	31,282.64	1086.485
500	122.928	28,524.97	31,881.26	1086.540
600	164.324	28,059.26	31,625.24	1086.377
700	209.893	26,659.32	30,449.29	1085.977
800	260.372	24,022.46	28,082.24	1085.264
900	317.788	19,196.73	23,668.35	1084.005
950	351.162	14,860.47	19,735.78	1082.898

Table IX. Results from Modified Catenary Theory for Figure 4, ϕ = .75 in., E = 10 x 10^6 p.s.i., $\bar{w}$ = 1.035 p.p.f., W = 4000 lbs., maximum B = 20,000 lbs., unstretched cable length = 1,079.1263 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)	Stretched cable length (ft)
50	-1.688	9,982.89	11,051.24	1081.814
100	7.092	12,487.70	13,807.20	1082.469
200	32.079	15,392.64	16,941.70	1083.195
300	62.623	17,007.64	18,693.80	1083.580
400	97.136	17,857.16	19,650.30	1083.769
500	135.070	18,111.52	20,000.00	1083.807
600	176.301	17,813.23	19,795.13	1083.708
700	221.010	16,909.54	18,991.85	1083.458
800	269.826	15,211.71	17,416.20	1083.014
900	324.535	12,144.92	14,538.21	1082.232
950	355.980	9,463.21	12,041.54	1081.561

Table X. Results from Modified Catenary Theory for Figure 4, ϕ = .75 in., E = 10 x 10^6 p.s.i., $\bar{w}$ = 1.035 p.p.f., W = 2000 lbs., unstretched cable length = 1,079.1263 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)	Stretched cable length (ft)
50	1.738	6,644.61	7,425.89	1080.931
100	11.715	8,178.41	9,067.18	1081.321
200	38.509	10,017.86	11,011.25	1081.772
300	70.282	11,056.49	12,113.71	1082.017
400	105.548	11,605.04	12,712.36	1082.138
500	143.798	11,768.59	12,920.68	1082.164
600	184.912	11,573.57	12,769.64	1082.103
700	229.061	10,989.82	12,233.25	1081.947
800	276.830	9,907.45	11,208.45	1081.670
900	329.922	8,008.39	9,396.80	1081.195
950	360.304	6,445.30	7,913.75	1080.807

Table XI. Results from Modified Catenary Theory for Figure 4, ϕ = .75 in., $E = 10 \times 10^6$ p.s.i., $\bar{w}$ = 1.035 p.p.f., W = 8000 lbs., unstretched cable length = 1,079.1263 ± .001 ft.

x_c (ft)	y_c (ft)	A (lbs)	B (lbs)	Stretched cable length (ft)
50	-5.549	16,008.46	17,513.62	1083.387
100	1.288	19,894.53	21,969.13	1084.446
200	23.449	24,365.54	26,956.80	1085.595
300	52.108	26,840.20	29,740.42	1086.196
400	85.472	28,137.80	31,279.30	1086.487
500	122.928	28,525.10	31,881.38	1086.540
600	164.328	28,060.67	31,626.63	1086.376
700	209.893	26,659.38	30,449.34	1085.977
800	260.362	24,018.05	28,077.93	1085.267
900	317.760	19,184.10	23,656.29	1084.013
950	351.127	14,842.20	19,718.88	1082.911

Table XII. Results from Modified Catenary Theory for Figure 3, ϕ = .75 in., T_0 = H = 20,000 lbs., $\bar{w}$ = 1.035 p.p.f. and E = 10 x 10^6 p.s.i.

x (ft)	θ (deg)	T (lbs)	y (ft)	s (ft)	$\bar{L}$ (ft)
0	0	20,000.000	0	0	0
100	.29517	20,000.265	.2576	100.0004	99.5498
250	.73790	20,001.659	1.6099	250.0069	248.8802
500	1.47568	20,006.635	6.4400	500.0553	497.8017
1,000	2.95039	20,026.546	25.7641	1,000.4424	995.9337
2,500	7.35892	20,166.103	161.2126	2,506.9171	2,495.6193
5,000	14.59792	20,667.171	647.5281	5,055.4749	5,032.6916
10,000	28.29178	22,713.197	2,633.3138	10,448.2366	10,401.1499

LIST OF FIGURES

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Figure 2.--Cable segment of stretched length ($s - s_0$)

Figure 3.--Cable extending outward from anchor at which $\theta_0 = 0$ and $T_0 = H$

Figure 4.--Cable supporting concentrated load, W

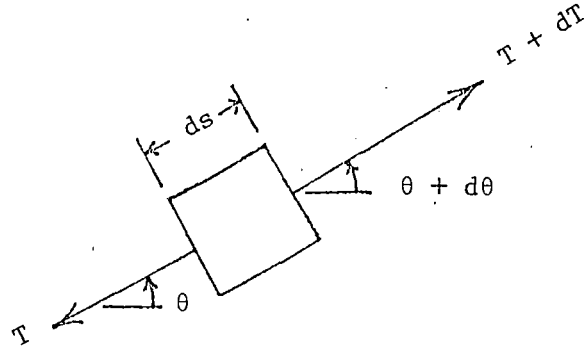

Figure 1. Cable element

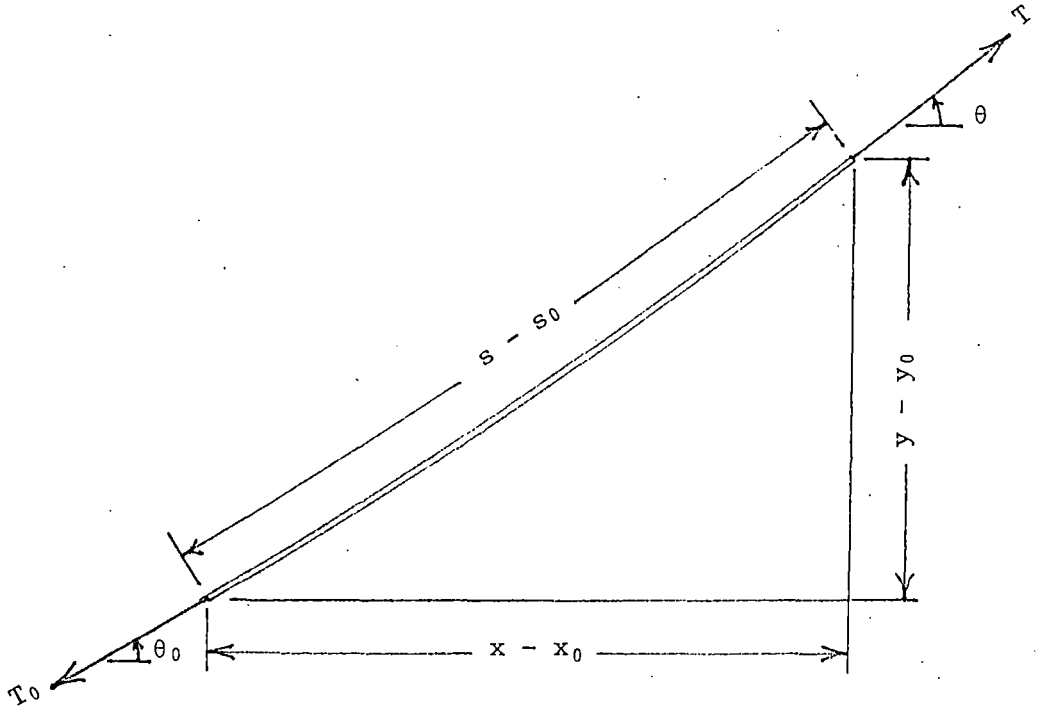

Figure 2. Cable segment of stretched length ($s - s_0$)

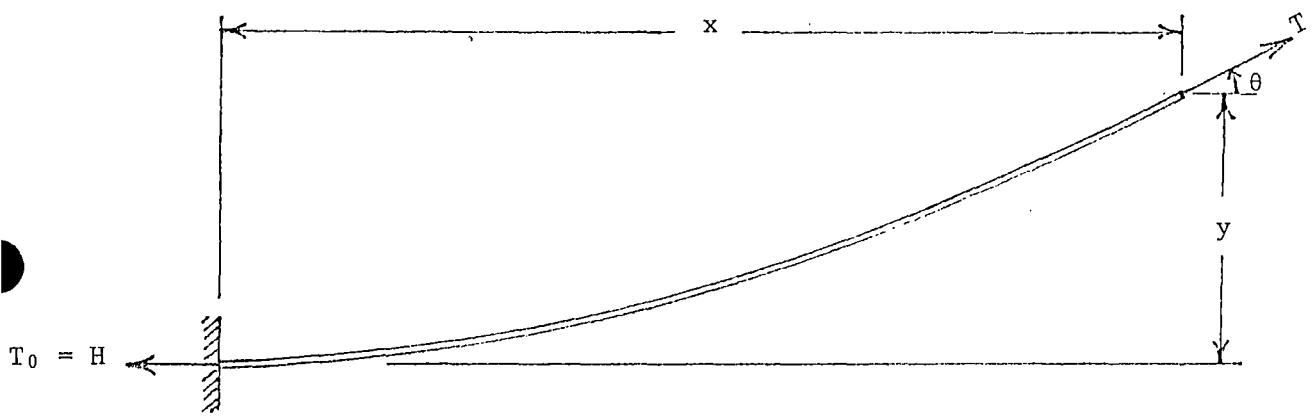

Figure 3. Cable extending outward from anchor at which $\theta_0 = 0$ and $T_0 = H$

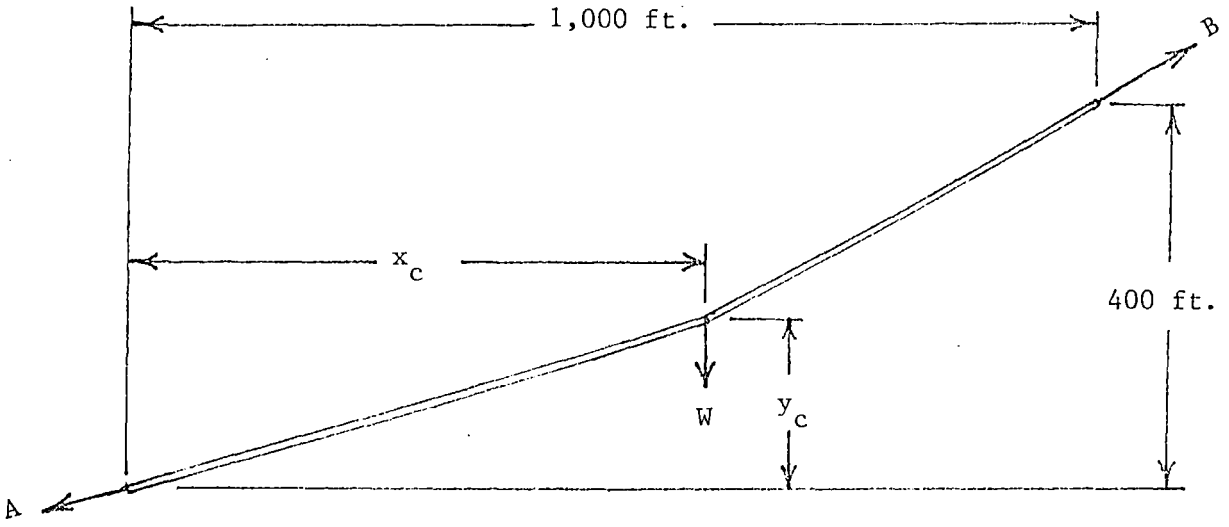

Figure 4. Cable supporting concentrated load, W.